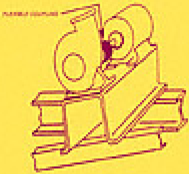
COUPLINGS AND JOINTS

DESIGN, SELECTION, AND APPLICATION

SECOND EDITION,
REVISED AND EXPANDED



JON R. MANCUSO

COUPLINGS AND JOINTS

DESIGN, SELECTION, AND APPLICATION

SECOND EDITION,
REVISED AND EXPANDED

JON R. MANCUSO

Kop-Flex
Emerson Power Transmission Corporation
Baltimore, Maryland

MARCEL DEKKER, INC.   NEW YORK • BASEL

ISBN: 0-8247-9950-X

This book is printed on acid-free paper.

Headquarters
Marcel Dekker, Inc.
270 Madison Avenue, New York, NY 10016
tel: 212-696-9000; fax: 212-685-4540

Eastern Hemisphere Distribution
Marcel Dekker AG
Hutgasse 4, Postfach 812, CH-4001 Basel, Switzerland
tel: 41-61-261-8482; fax: 41-61-261-8896

World Wide Web
http://www.dekker.com

The publisher offers discounts on this book when ordered in bulk quantities. For more information, write to Special Sales/Professional Marketing at the headquarters address above.

Copyright © 1999 by Marcel Dekker, Inc. All Rights Reserved.

Neither this book nor any part may be reproduced or transmitted in any form or by any means, electronic or mechanical, including photocopying, microfilming, and recording, or by any information storage and retrieval system, without permission in writing from the publisher.

Current printing (last digit):
10 9 8 7 6 5 4 3 2 1

PRINTED IN THE UNITED STATES OF AMERICA

In memory of

Lynn—my daughter
Samuel—my father

and to the rest of my family

Josephine—my mother
Rose—my wife
Samuel, Jonette, and Kristie—my children

and the next generation, my grandchildren

Jonathan
Lynnette
Allec
Brandon
D.J.

Preface to the Second Edition

The first edition of *Couplings and Joints* was published in 1986 and it took me 5 years to put it together. When I started work on this edition, I thought the task would be a lot simpler, but its preparation has taken me almost as long. The new edition is almost 100 pages longer. It has over 700 illustrations and tables (over 150 more than in the first edition) to help the reader understand coupling design, selection, and application.

The first chapter updates the history of couplings since the first edition. Chapter 2 has been expanded to give more detail on the types of couplings, how they work, and where they are used. Here I have classified couplings into general-purpose couplings and special-purpose couplings, and the chapter discusses the types and usage in each of these categories. Tables have been updated for capacities and characteristics and broken down into general-purpose couplings and special-purpose couplings. Chapter 4 has been expanded and provides the latest information on alignment, including laser alignment and signature analysis. Chapter 6 has been greatly expanded to include some of the latest developments in gear couplings, such as carburized ground spindles. An updated table on $S_C V$ criteria for gear couplings is included, giving additional materials and information on heat treatment. The section on U-joints now offers more details on the various types of industrial U-joints and many examples of the various types of failures seen by U-joints.

Chapter 8 has been expanded to include several methods for calculating coupling guard temperatures. Guidelines on how to design a guard that will not overheat are also given. Metallic element couplings have greatly improved since the first edition. They have become the preferred choice for new equipment, because they require less maintenance than lubricated couplings. Because of improvements in materials and the manufacturing process, the reliability of these couplings has greatly increased.

A new chapter has been added, Chapter 10, which covers some of the topics most commonly discussed when rotating engineers get together. This chapter covers torquemeter couplings, why they are used, and the advantages and disadvantages of various types. There is a section in this chapter on torsional damping couplings: what they do and the various types available. Also in Chapter 10 is a section on the most discussed topic at technical meetings, the replacement of lubricated couplings with nonlubricated couplings. The section covers what needs to be considered, the pros and cons. It also gives guidelines for replacements. Also given is an example for replacing gear couplings with diaphragm couplings for a high-speed drive train.

Chapter 11 presents coupling nomenclature; these terms will be found in catalogues, in specifications, and in the discussions in this book.

I thought in 1986 that I had written everything on couplings that there was to be written. But in more than 30 years of designing, selecting, and applying couplings, I continue to find that I learn something new about them almost every day.

Jon R. Mancuso

Preface to the First Edition

The flexible coupling method of connecting rotating equipment is a vital and necessary technique. Large shafts in loosely mounted bearings, bolted together by flanged rigid couplings, do not provide for efficient and reliable mechanical power transmission. This is especially true in today's industrial environment, where equipment system designers are demanding higher speeds, higher torques, greater flexibility, additional misalignment, and lighter weights for flexible couplings. The need for flexible couplings is becoming more acute as is the need for technological improvements in them.

The basic function of a coupling is to transmit torque from the driver to the driven piece of rotating equipment. Clutches (which constitute another subject for a book and are not covered here) are couplings designed to cease transmitting torque at a certain load or speed. Flexible couplings expand upon the basic function by also accommodating misalignment and end movement. During the initial assembly and installation of rotating equipment, precise alignment of the equipment shaft axes is not only difficult to achieve, but in most cases it is economically prohibitive. In addition, misalignment during equipment operation is even more difficult to control. Misalignment in operation can be caused by flexure of structures under load, settling of foundations, thermal expansion of equipment and their supporting structures, piping expansion or contraction, and many other factors. A flexible coupling serves as a means to compensate for, or minimize the effects of, misalignment. Flexible couplings, however, have their own limitations. Therefore, calculations and predictions are required to know what the maximum excursions can be. Only then can the correct coupling be selected.

A system designer or coupling user cannot just put any flexible coupling into a system and hope it will work. It is the responsibility of the designer or user to select a compatible coupling for the system. The designer must also be the coupling selector. Flexible coupling manufacturers are not system designers and should not be expected to assume the role of coupling selectors. Since they are not system experts, they can only take the information given them and size, design, and manufacture a coupling to fulfill the requirements specified. Some coupling manufacturers can offer some assistance but they usually will not accept responsibility for a system's successful operation.

The purpose of this book is to aid the coupling selector (system designer or user) in understanding flexible couplings so the best coupling can be selected for each system. In order to help coupling selectors do their jobs properly, four chapters are included that cover the basics of couplings. These chapters provide answers to often-asked questions, such as:

Why a flexible coupling?
What are this coupling's limitations and capabilities?
How much unbalance will this coupling produce?
How can I compare this coupling with that coupling?
How do I install this coupling?
How is it disassembled?
How is it aligned?
How is it lubricated?

There are two basic classes of couplings: the rigid coupling and the flexible coupling. A chapter on rigid couplings has been included. Rigid couplings should only be used when the connecting structures and equipment are rigid enough so that very little misalignment can occur and the equipment is strong enough to accept the generated moments and forces.

There are hundreds of different types of flexible couplings. In this book, many types are covered in general and the more common ones in detail. The eight most common types of couplings in use today are:

Gear couplings
Grid couplings

Chain couplings
Universal joints
Elastomeric shear couplings
Elastomeric compression couplings
Disk couplings
Diaphragm couplings

Each one of these couplings is covered in detail. Included in each section are variations available, the principles of operation, coupling constructions, design criteria, failure modes, and other important information.

It is hoped that this book will achieve two things: generally, to stress to the reader the importance of couplings in power transmission systems; and, more specifically, to provide the coupling selector with the basic tools required for the successful application of couplings to particular needs.

This book could not have been written without the help and cooperation of many individuals and coupling manufacturers. In particular, acknowledgment is given to the Mechanical Drives Division of Zurn Industries, Inc. and especially to the following personnel: Norman Anderson, Sam Steiner, Jim Paluh, Bill Herbstritt, and my secretary Marty Keim. Special thanks are extended to: Edward Heubel and Bill Herbstritt, for many of the sketches; Michael M. Calistrat of Boyce Engineering, Howard Schwerdlin of Lovejoy, and Q. W. Hein of the Falk Corporation, for their extra help.

Jon R. Mancuso

Contents

Preface to the Second Edition *v*
Preface to the First Edition *vii*

1. History of Couplings **1**

- I. Early History 1
- II. The Period 1990–1930 2
- III. The Period 1930–1945 4
- IV. The Period 1945–1960 10
- V. The Period 1960–1985 10
- VI. The Period from 1985 to the Present 15
- VII. Conclusion 15

2. Overview of Couplings and Joints **23**

- I. Advantages of Using Flexible Couplings 23
- II. Types of Flexible Couplings—General 29
- III. Types of General-Purpose Couplings 34

IV. Special-Purpose Couplings 52
V. Characteristics and Capacities of Couplings 67

3. Selection and Design **75**

I. Selection Factors 75
II. Design Equations and Parameters 80
III. Balancing of Couplings 118
IV. Interface Connections 152

4. Installation and Maintenance **177**

I. Introduction 177
II. Installation of Couplings 178
III. Alignment of Equipment 193
IV. Coupling Assembly 215
V. Lubrication of Couplings 227
VI. Coupling Disassembly 240

5. Rigid Couplings **247**

I. Introduction 247
II. Flanged Rigid Couplings 249
III. Ribbed Rigid Couplings 249
IV. Sleeve Rigid Couplings 253
V. Quill Shaft Rigid Couplings 258

6. Mechanically Flexible Couplings **261**

I. Introduction 261
II. Gear Couplings 262
III. Grid Couplings 343
IV. Chain Couplings 354
V. Universal Joints 362

7. Elastomeric Couplings **403**

I. Introduction 403
II. Elastomeric Compression Couplings 423
III. Elastomeric Shear Couplings 430

8. Metallic Element Couplings **439**

I. Introduction 439

II. Disc Couplings 462
III. Diaphragm Couplings 475

9. Miscellaneous Couplings 489

I. Introduction 489
II. Pin and Bushing Coupling 490
III. Offset Coupling (Schmidt Coupling) 491
IV. Tangential Spring Coupling 491
V. Metallic Beam Coupling 492
VI. Sliding Block Coupling 493

10. Special-Purpose Couplings 495

I. Torquemeter Couplings 495
II. Torsional Damping Couplings 508
III. Retrofitting Nonlubricated Couplings 519

11. Coupling Nomenclature 537

Appendix A: Coupling Manufacturers 553
Appendix B: Typical Installation Instructions 563
References 569
Index 573

COUPLINGS AND JOINTS

1

History of Couplings

I. EARLY HISTORY

The flexible coupling is an outgrowth of the wheel. In fact, without the wheel and its development there would have been no need for flexible couplings. It has been reported that the first wheel was made by an unknown Sumerian more than 5000 years ago in the region of the Tigris and Euphrates rivers. The earliest record we have dates to 3500 B.C. History records that the first flexible coupling was the universal joint (see Figure 1.1), used by the Greeks around 300 B.C. The Chinese were using this concept sometime around A.D. 25. The father of the modern flexible coupling was Jerome Cardan, who in the sixteenth century invented what was described as a simple device consisting of two yokes, a cross, and four bearings. This joint, the common ancestor of all flexible couplings, is still in use today and is continually being upgraded with the latest technology. Cardan did not design the Cardan shaft for rotating-shaft applications, but only as a suspension member. The Cardan joint

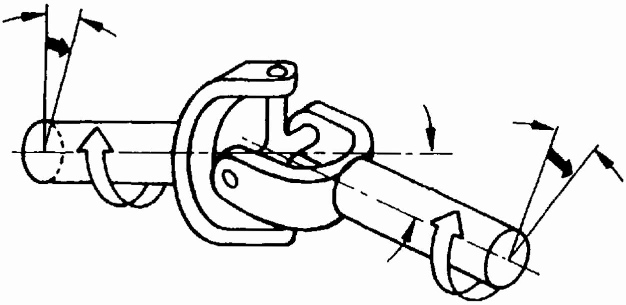


FIGURE 1.1 The ancestor of the flexible coupling—the universal joint.

is also known as the Hooke joint. In approximately 1650, Robert Hooke developed the first application of this joint, using it in a rotating shaft in a clock drive. Hooke wrote the equation for the fluctuations in angular velocity caused by a single Cardan joint.

Between 1700 and 1800, very little is recorded in the way of further developments in flexible couplings. The Industrial Revolution and especially, later, the automobile revolution precipitated the creation of many flexible couplings. In 1886, F. Roots theorized that if he thinned down the flange section of a rigid coupling it would flex and prevent the equipment and shaft from failing, an idea that was the forerunner of today's diaphragm coupling (see Figure 1.2A). The Davis compression coupling (Figure 1.2B) was developed to eliminate keys by compressing hubs onto the shaft. It was thought to be safer than other coupling devices because no protruding screws were required. What is believed to be the first chain coupling (Figure 1.2C) was described in the May 1914 issue of *Scientific American.*

II. THE PERIOD 1900–1930

The coupling industry expanded rapidly in the 1920s as a direct result of the invention of the automobile. Table 1.1 gives some of the coupling manufacturers established during this period. (See Figures 1.3A–H.)

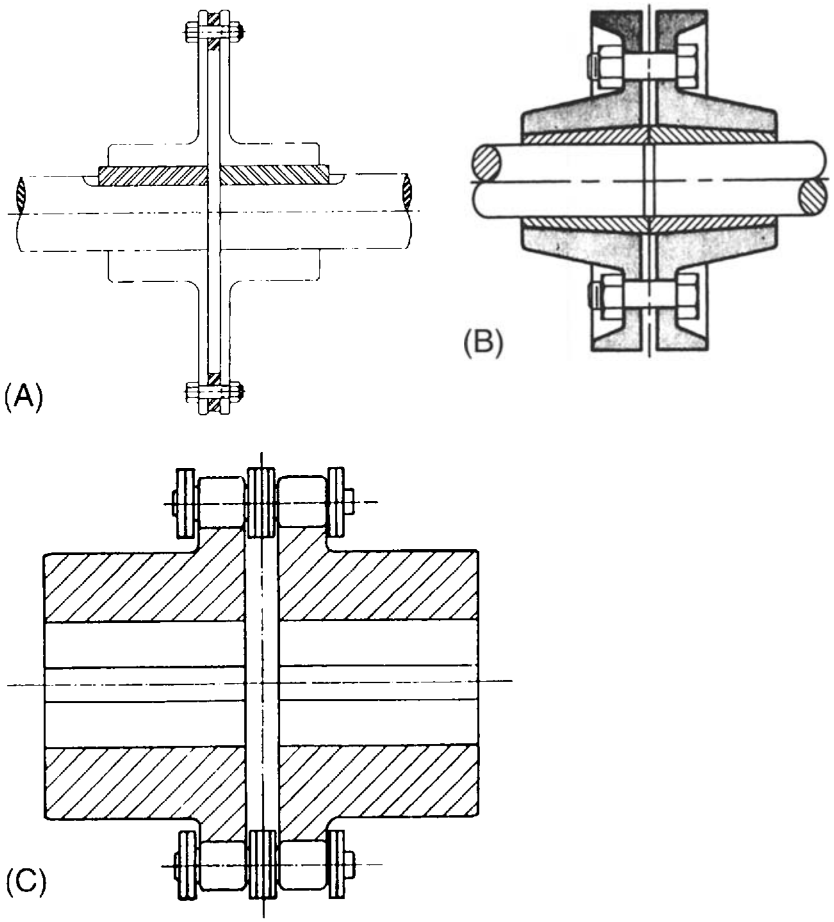


Figure 1.2 Early couplings. (A) Diaphragm coupling patented in 1886. (B) Davis compression coupling. (C) Early chain coupling.

TABLE 1.1 Coupling Manufacturers Established in the 1920s

Original name	Date started	Today's name	Figure number
Thomas Flexible Coupling Company	1916	Thomas Flexible Disc Coupling Operation of Rexnord Corporation	1.3A
Fast's Couplings, The Barlette Hayward Co.	1919	Kop-Flex, Inc., Emerson Power Transmission Corporation	1.3B
Lovejoy Couplings	1927	Lovejoy, Inc.	1.3C
Poole Foundry and Machine	1920	TB Wood's, Inc.	1.3D
American Flexible Couplings	1928	Ameridrives International	1.3E
Ajax Flexible Coupling Company, Inc.	1920	Renold, Inc., Engineered Products	1.3F
TB Wood's & Sons, Inc.	1920	TB Wood's, Inc.	1.3F
Bibby	1919	Bibby	1.3H

III. THE PERIOD 1930–1945

During the period 1930–1945 many general-purpose flexible couplings were introduced into the industrial market. The most frequently used are illustrated in Figure 1.4:

Chain coupling (Figure 1.4A)
Grid coupling (Figure 1.41B)
Jaw coupling (Figure 1.4C)
Gear coupling (Figure 1.4D)
Disc coupling (Figure 1.4E)
Slider block coupling (Figure 1.4F)
Universal joint (Figure 1.4G)

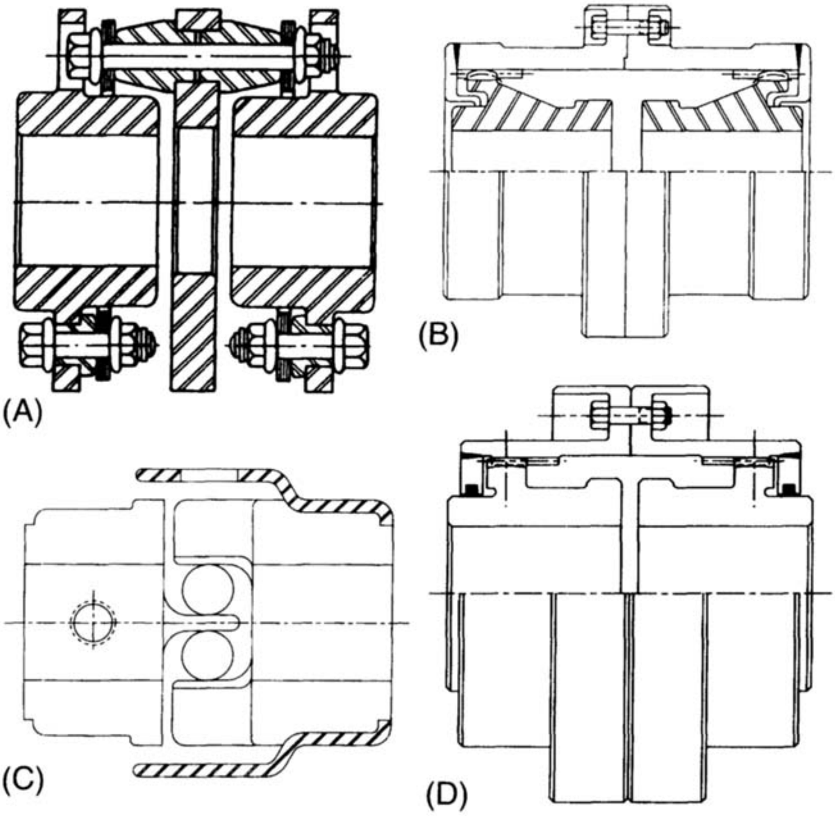


Figure 1.3 Couplings of the period 1900–1930. (A) Thomas disc coupling. (B) Fasi's gear coupling. (C) Lovejoy jaw coupling. (D) Poole gear coupling. (E) American slider block coupling. (F) Ajax pin-and-bushing coupling. (G) TB Wood's resilient coupling. (H) Bibby grid coupling.

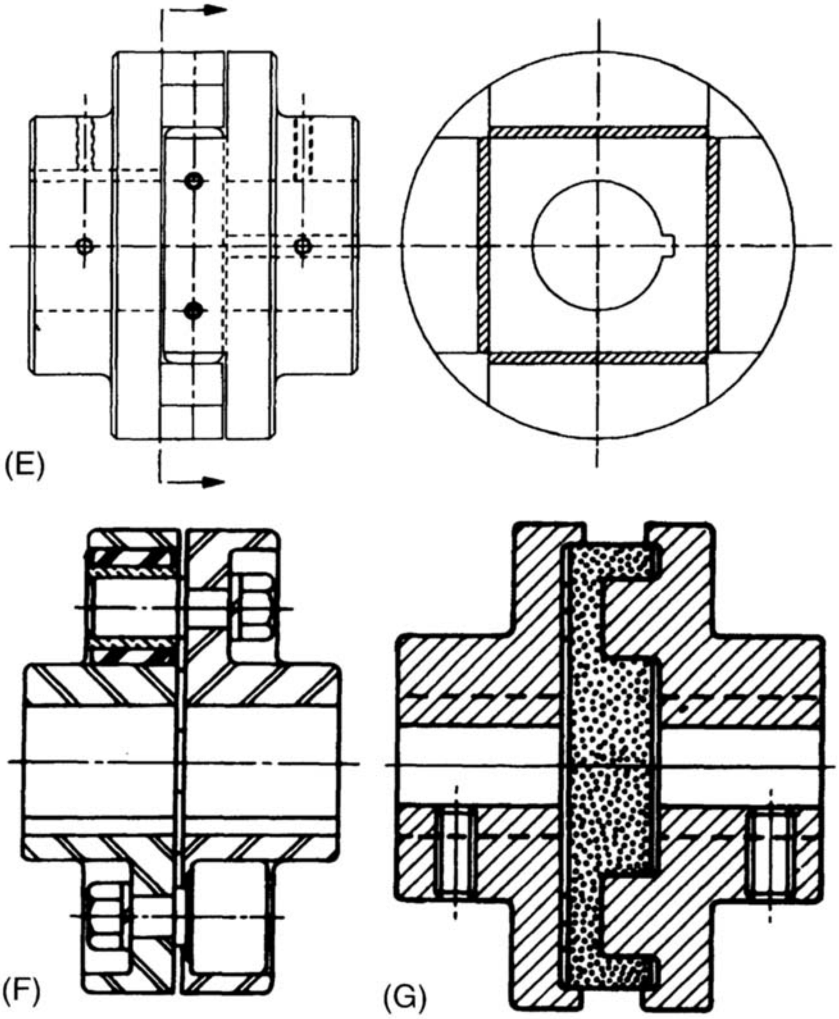
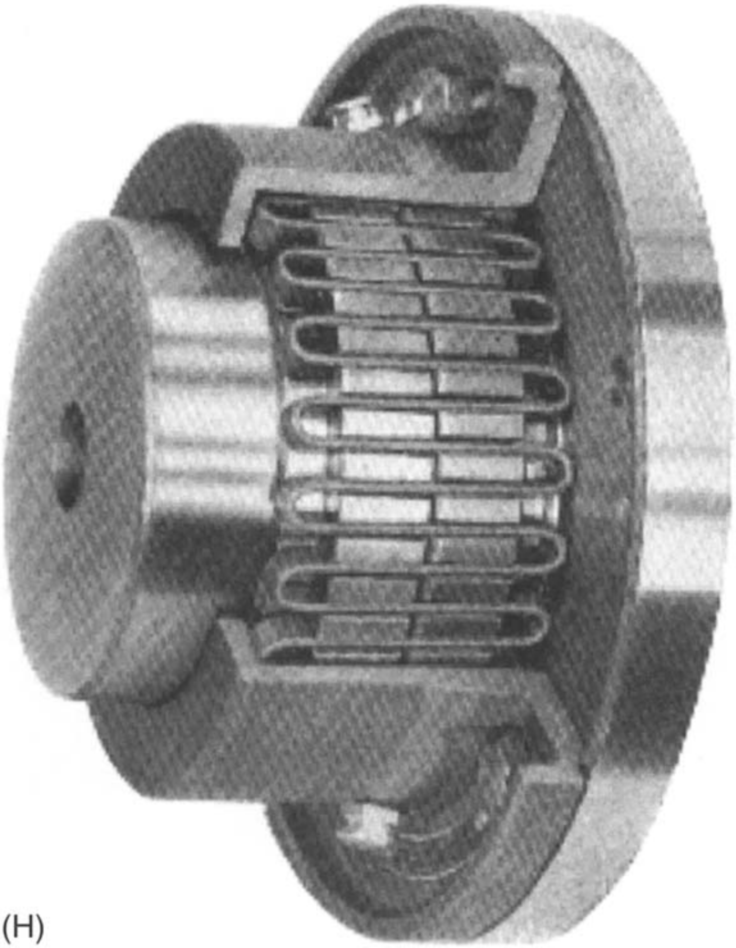


FIGURE 1.3 Continued.



(H)

Figure 1.3 Continued.

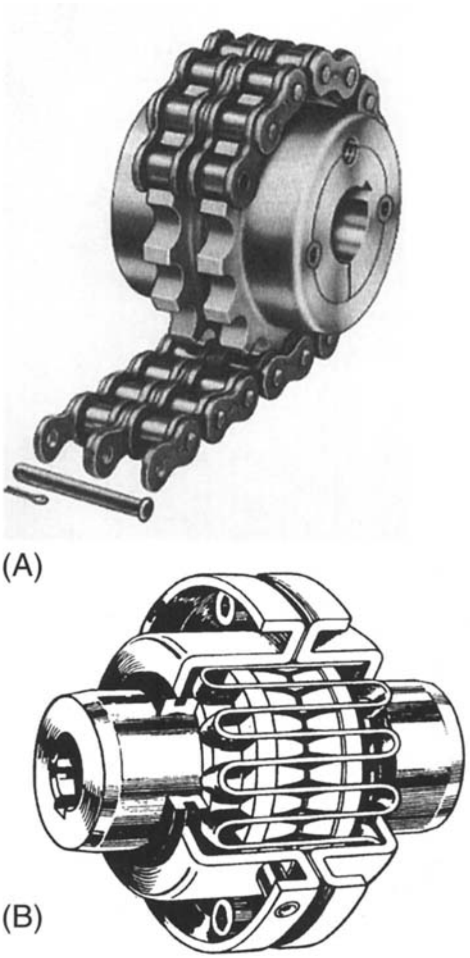


Figure 1.4 Couplings of the period 1930–1945. (A) Chain coupling (courtesy of Dodge Division of Reliance Electric). (B) Grid coupling (courtesy of Falk Corporation). (C) Jaw coupling. (D) Gear coupling. (E) Disc coupling (courtesy of Coupling Division of Rexnord, Thomas Couplings). (F) Sliding block coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (G) Universal joint (courtesy of Spicer Universal Joint Division of Dana Corporation).

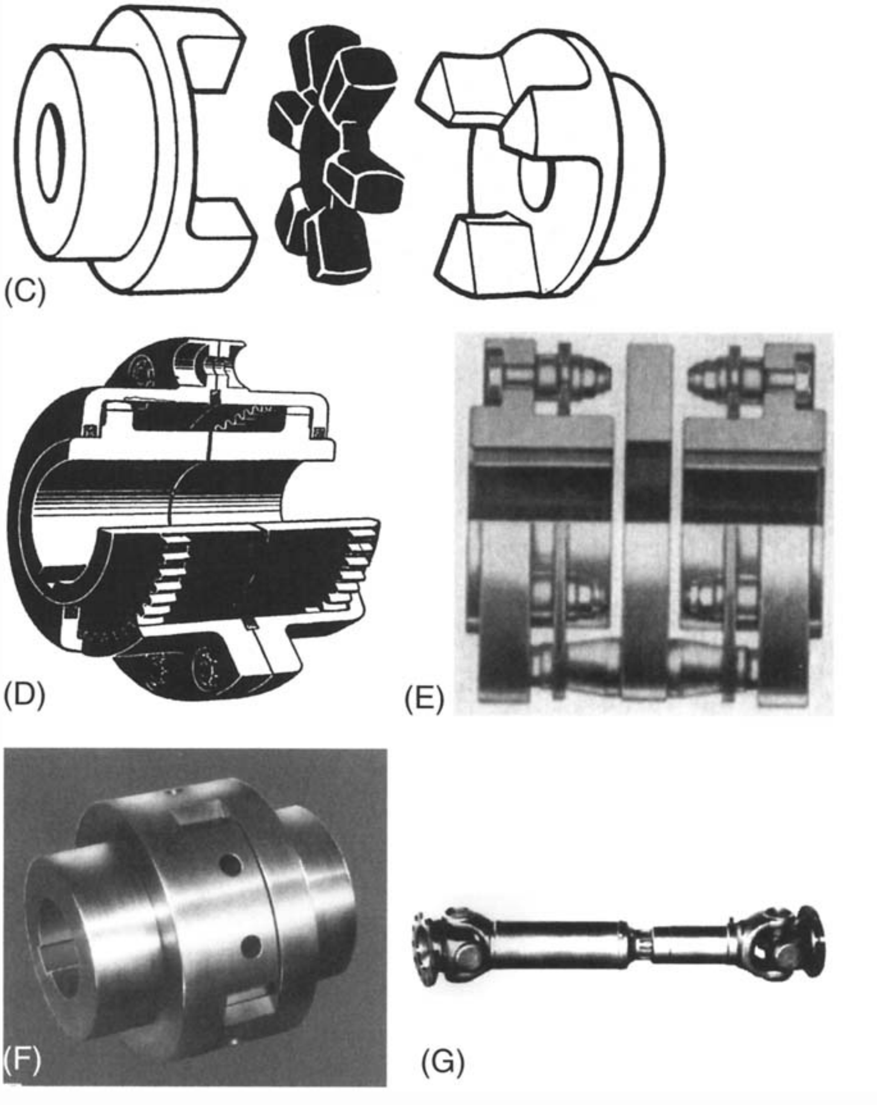


Figure 1.4 Continued.

IV. THE PERIOD 1945–1960

The late 1940s through the 1950s saw rapid technological advancement and the introduction of rotating equipment. Larger and higher horsepower equipment came into use. This brought about the need for more "power-dense" flexible couplings with greater misalignment, which had to be accommodated.

Around this time the fully crowned gear spindle (Figure 1.5A) was developed and introduced into the steel industry. Also during this period, the use of the gas turbine in industrial applications (generators, compressors) was becoming popular. With increased use of the gas turbine came the requirement for higher speed couplings. Therefore, the gear coupling and disc coupling were upgraded and improved to handle higher speed requirements (see Figures 1.5B and 1.5C).

Rotating equipment demands higher and higher operating speeds. With the increased speed of operation came system problems that required lighter weight couplings, and the torsional characteristics of couplings became more important. This necessitated improvements in resilient couplings (see Figure 1.5D), as these couplings not only had to help tune a system but in many cases had to be able to absorb (dampen) anticipated peak loads caused by torsional excitations.

V. THE PERIOD 1960–1985

Higher horsepower and higher speed continued to be increasing requirements of rotating equipment. The 1960s saw the introduction of many new types of couplings. Many coupling manufacturers introduced a standard line of crowned tooth gear couplings (see Figure 1.6A). The gear coupling was probably the most widely used type of coupling on the market. A half dozen major manufacturers produced virtually interchangeable product lines.

The grid coupling and chain coupling were also very popular for general-purpose applications (see Figures 1.6B and 1.6C). The rubber tire coupling (Figure 1.6D) was widely used, with many companies offering a model. More sophisticated resilient couplings (Figures 1.6E and 1.6F) were introduced during this period to help with the ever-

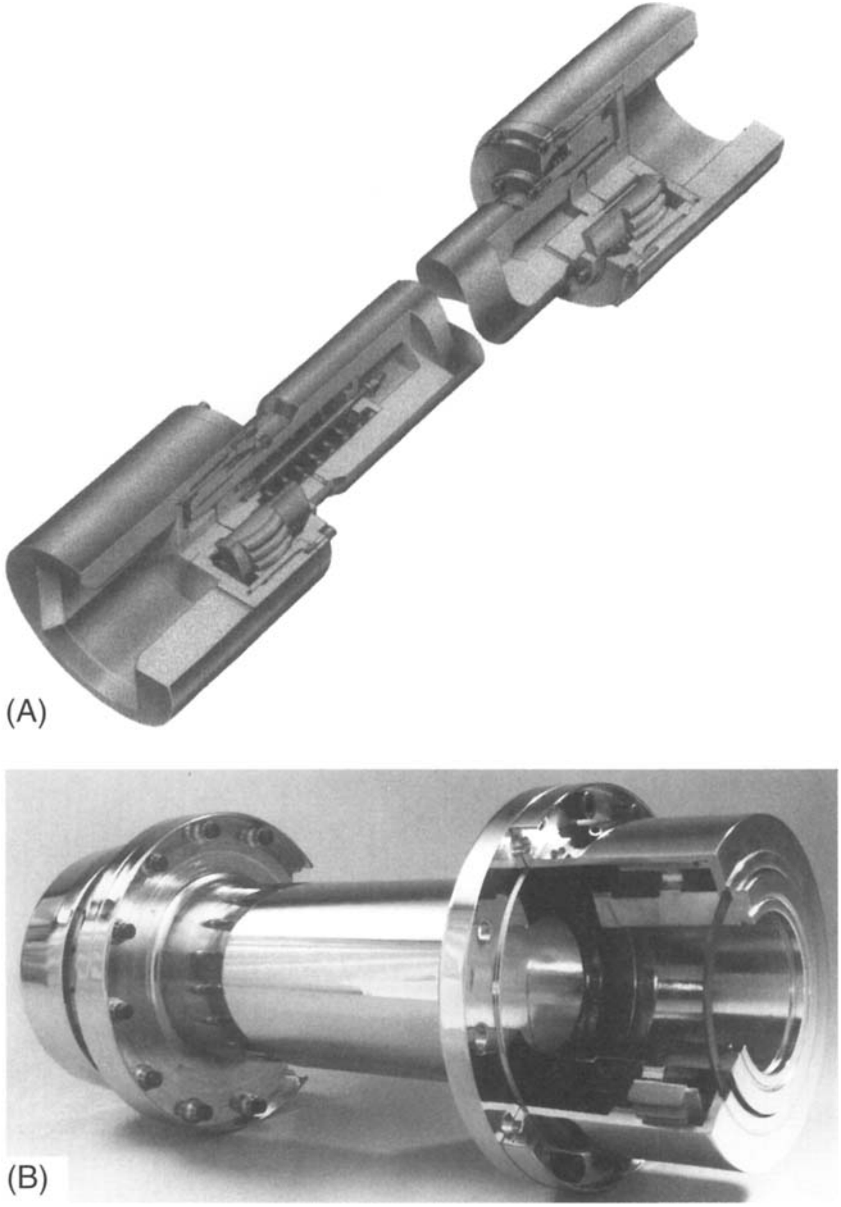


FIGURE 1.5 Couplings of the period 1945–1960. (A) Gear spindle coupling (courtesy of Renolds Industrial Contractors, Inc.). (B) High-speed gear coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (C) High-speed disc coupling (courtesy of Coupling Division of Rexnord, Thomas Couplings). (D) Resilent coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation).

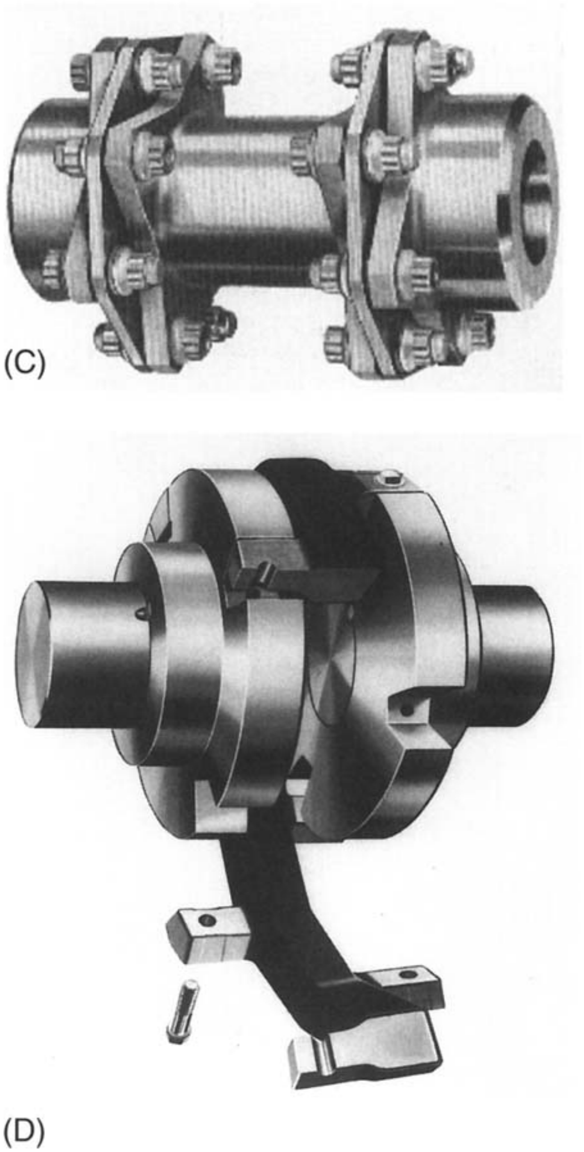


Figure 1.5 Continued.

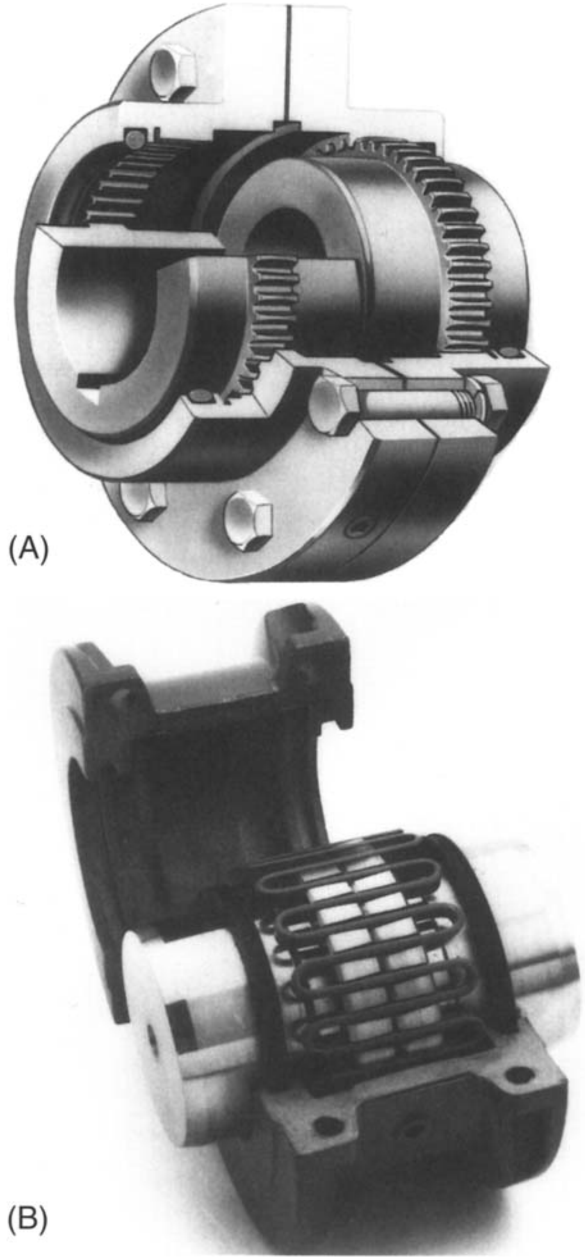


FIGURE 1.6 (A) Crowned tooth gear coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (B) Grid coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation).

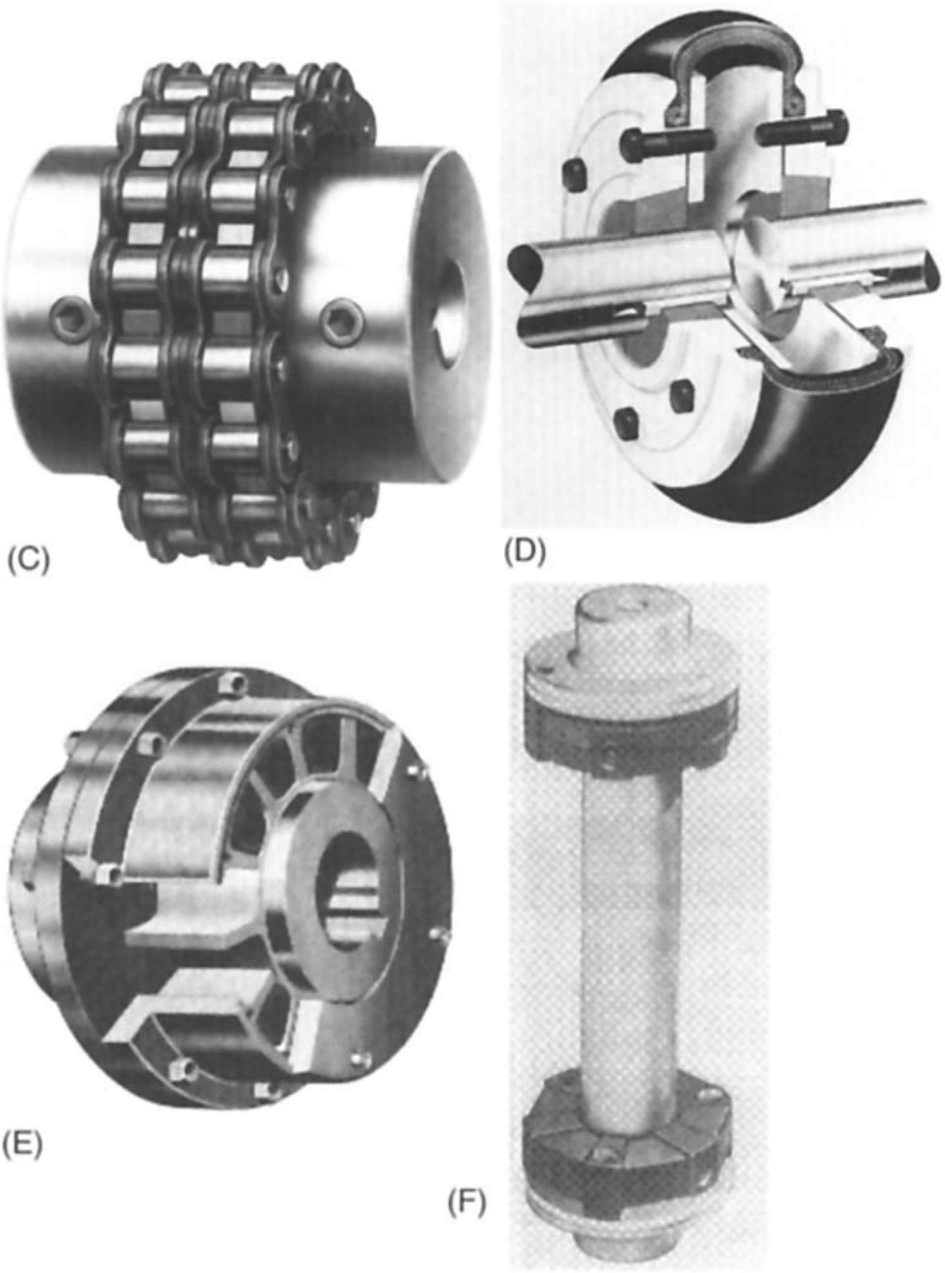


FIGURE 1.6 Continued. (C) Chain coupling (courtesy of Morse Industrial, Emerson Power Transmission Corporation). (D) Rubber tire coupling (courtesy of Dodge Division of Reliance Electric). (E) Elastomeric coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (F) Elastomeric coupling (courtesy of Lovejoy, Inc).

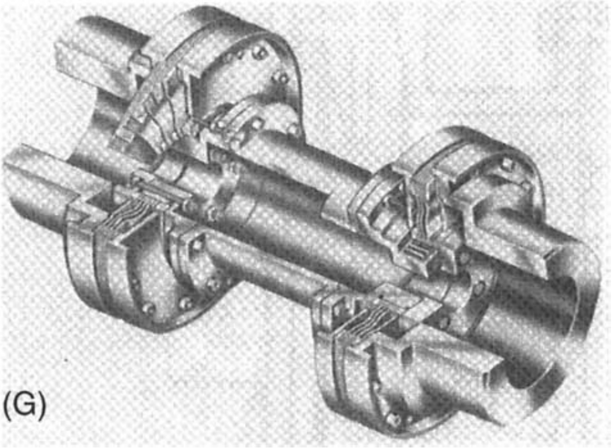
increasing system problems, as were several metallic element couplings (disc or diaphragm) (Figures 1.6G–I). The application of nonlubricated couplings grew rapidly during this time period. The gear coupling was upgraded to meet the challenge of higher speeds (Figure 1.6J), as was the ancestor of the flexible coupling, the universal joint (Figure 1.6K).

VI. THE PERIOD FROM 1985 TO THE PRESENT

The advancement in couplings and joints continues today, driven by the use of improved materials, finite element analysis (FEA), and new manufacturing methods. Nonlubricated couplings using FEA in their design are more reliable and have greater capacities (see Figure 1.7A). Advances in computer numerical control (CNC) equipment have eliminated the need for electron beam welding, bringing about the development of one-piece diaphragm couplings (see Figure 1.7B) and also the integral filler multiple diaphragm (see Figure 1.7C). Design optimization of resilient block shapes and improvements in resilient materials allow increased capacity and longer life of resilient couplings (see Figure 1.7D). Through FEA optimization and material optimization, U-joints have been developed that can handle more power in the same envelope (see Figure 1.7E). The demand for more efficient rotating equipment has created the need to monitor output torque, which has in turn brought about the development of several torque-monitoring couplings (see Figure 1.7F).

VII. CONCLUSION

Advances in couplings and universal joints are a continuing process as the needs of power transmission increase. In Chapter 2 we describe and discuss the types of couplings available today. Up to now we have been discussing relatively large couplings for industrial applications, generally with bores larger than 1/2 in. But there is another type of flexible coupling in widespread use: the miniature coupling used to drive servomechanisms, office equipment, and other small mechanisms. These miniature couplings will be discussed in Chapter 2.



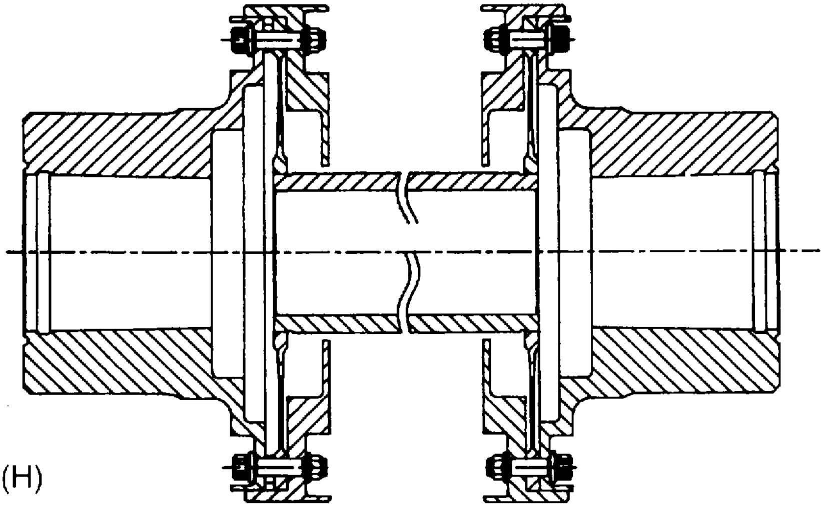


Figure 1.6 Continued. (G) Multiple convoluted diaphragm coupling (courtesy of Zurn Industries, Inc., Mechanical Drives Division). (H) Tapered contoured diaphragm coupling (courtesy of Lucas Aerospace Power Transmission Corporation). (I) Multiple diaphragm coupling (courtesy of Flexibox International, Metastream Couplings). (J) High-speed lightweight coupling (courtesy of Zurn Industries, Inc., Mechanical Drives Division). (K) Industrial universal joint (courtesy of Voithe Transmit GmbH).

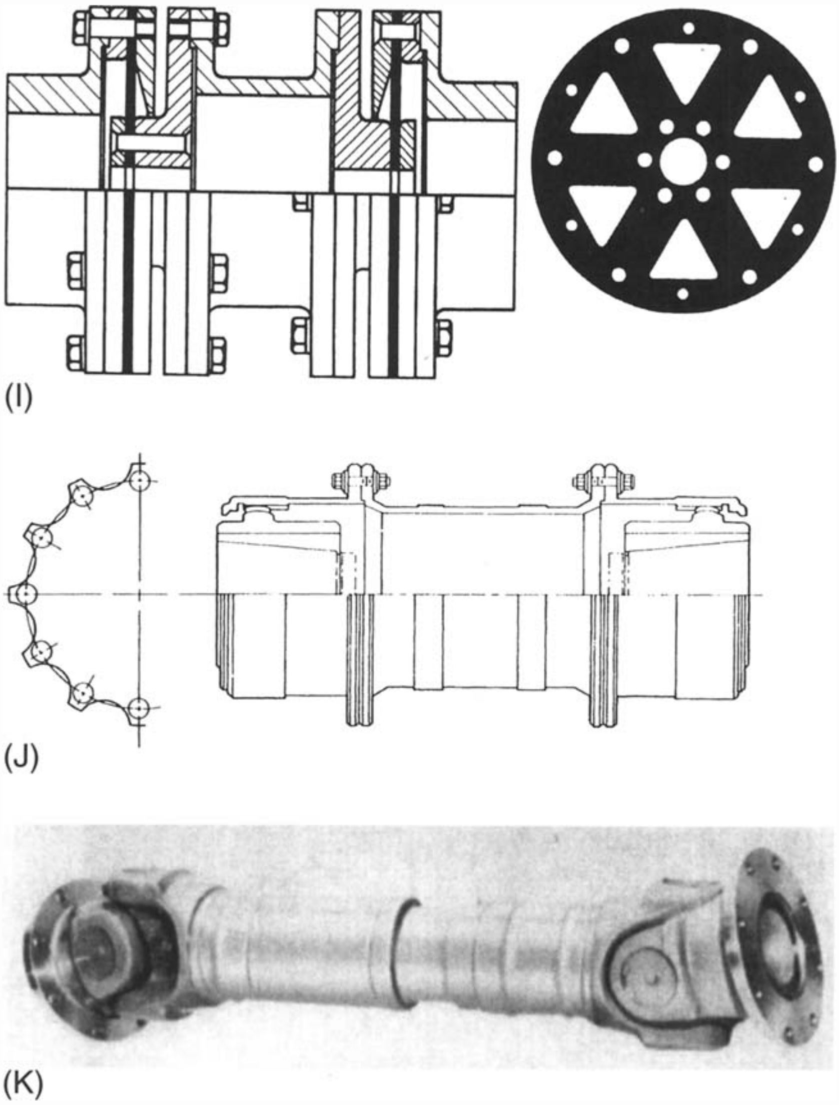
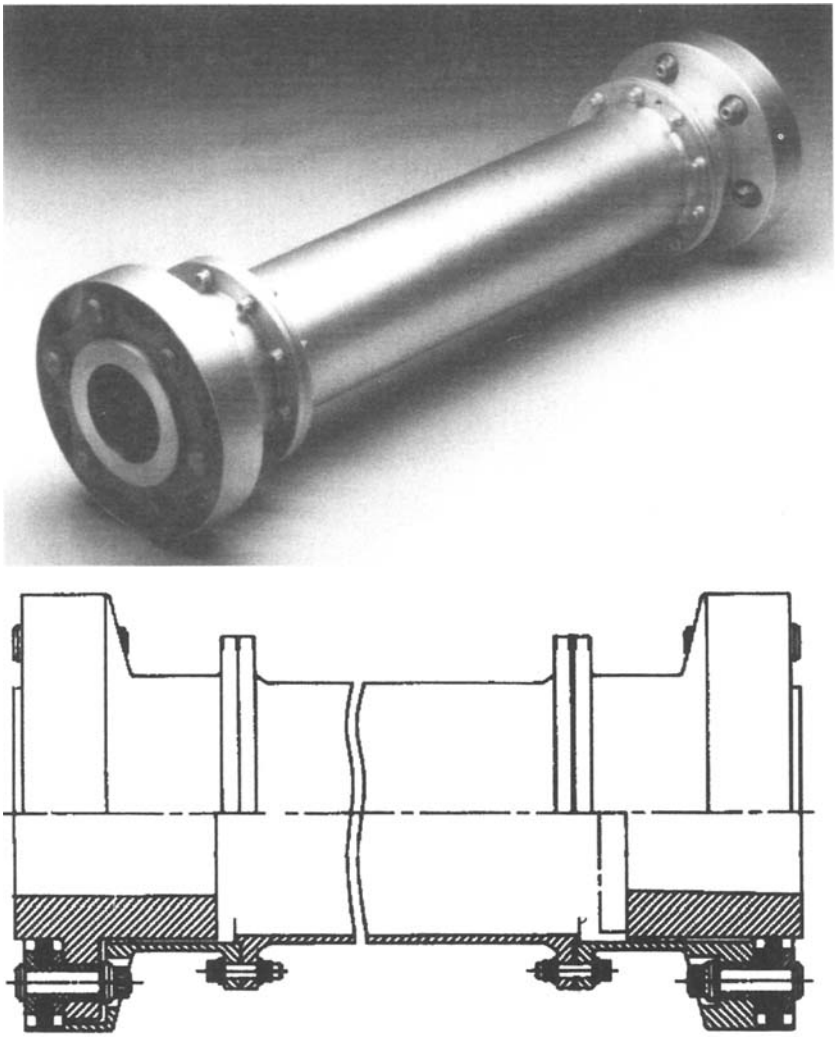


Figure 1.6 Continued.



(A)

FIGURE 1.7 (A) High-performance disc coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (B) One-piece diaphragm coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (C) Integral filler multiple diaphragm coupling (courtesy of Ameridrives International). (D) High-capacity elastomeric block coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation). (E) Block-type universal joint (courtesy of Kop-Flex Inc. Emerson Power Transmission Corporation). (F) Monopole torque-measuring coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation).

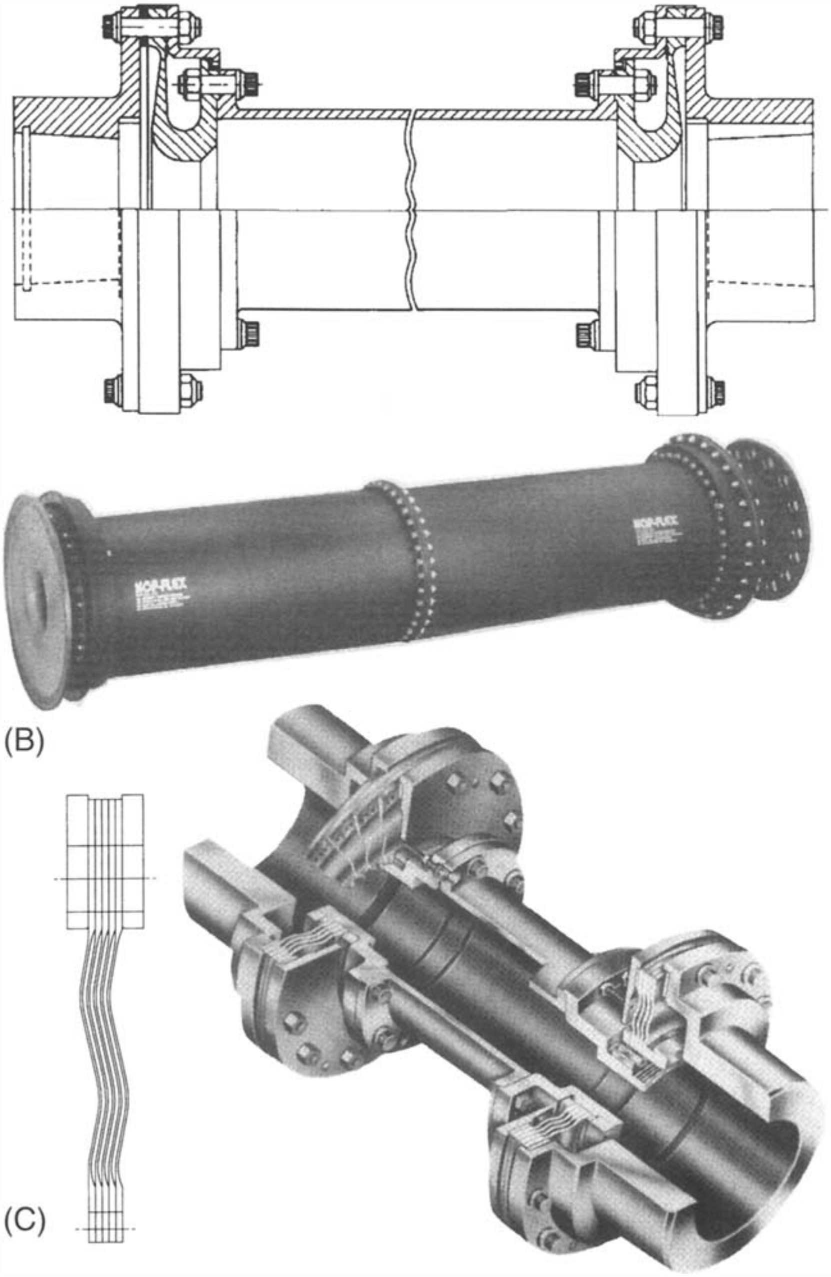


FIGURE 1.7 Continued.

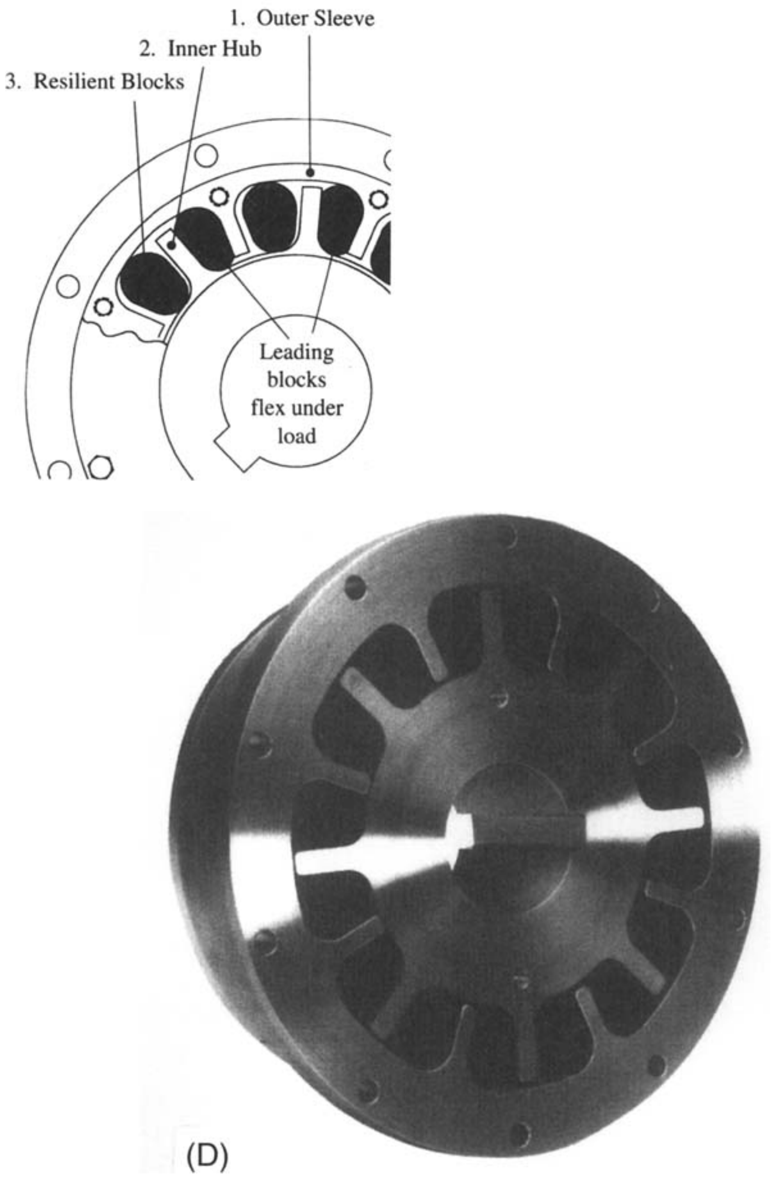


FIGURE 1.7 Continued.

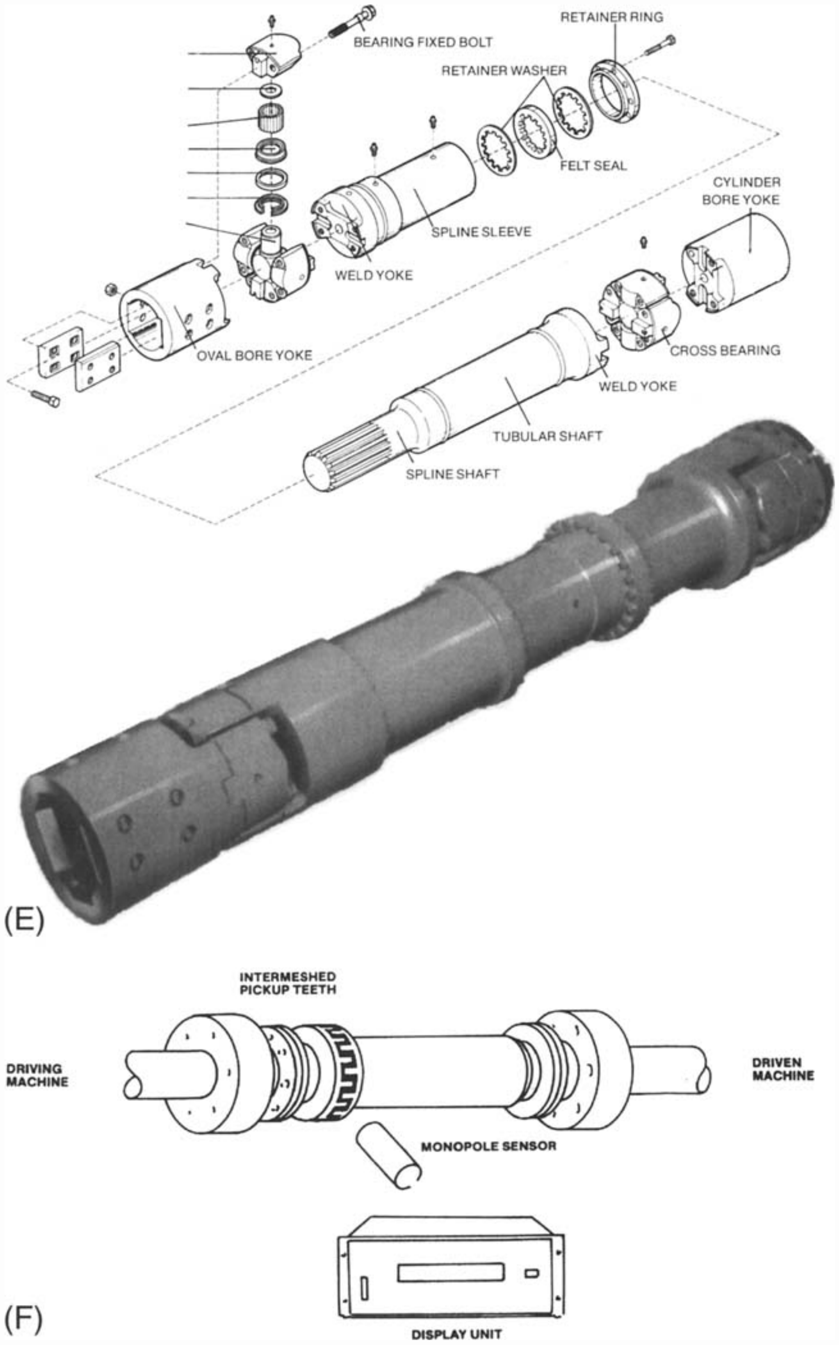


FIGURE 1.7 Continued.

Many of the flexible couplings used today will be around for years. As with the ancestor of flexible couplings, the universal joint, technological improvements in materials, design (such as FEA), and manufacturing will help upgrade couplings so they can handle the ever-increasing needs and demands of power transmission equipment.

2

Overview of Couplings and Joints

I. ADVANTAGES OF USING FLEXIBLE COUPLINGS

Historically, rotating equipment was first connected by means of rigid flanges (Figure 2.1). Experience indicates that this method did not accommodate the motions and excursions experienced by the equipment. As discussed in Chapter 1, F. Roots was the first to thin these flanges and allow them to flex. Rigid couplings are used to connect equipment that experiences very small shaft excursions or with shafts made long and slender enough that they can accept forces and moments produced from flexing flanges and shafts.

Flexible couplings join two pieces of rotating equipment while permitting some degree of misalignment or end movement or both. The three basic functions of a flexible coupling are to (Figure 2.2)

1. Transmit power (Figure 2.2A)
2. Accommodate misalignment (Figure 2.2B)
3. Compensate for end movement (Figure 2.3C)

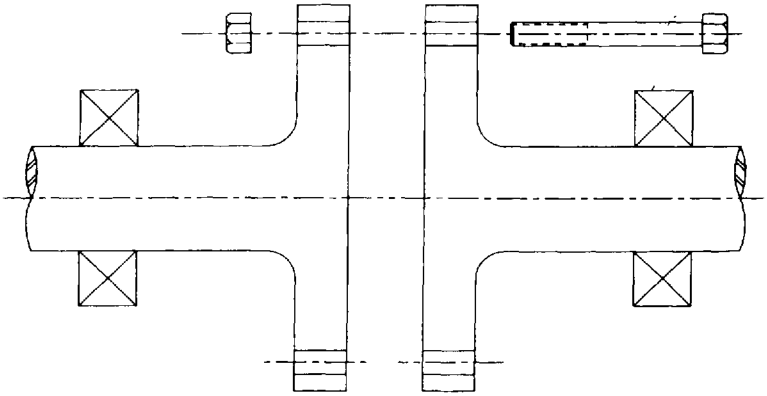


FIGURE 2.1 Rigid flanged connection.

A. Transmit Power

Couplings are primarily used to transmit mechanical power from one machine to another. The power is in the form of mechanical torque at some speed or work per unit of time. In general, the amount of power lost by a flexible coupling is small, although some couplings are more efficient than others.

B. Accommodate Misalignment

Flexible couplings must accommodate three types of misalignment (Figure 2.3):

1. Parallel offset. The axes of connected shafts are parallel but not in the same straight line (Figure 2.3A).
2. Angular. The axes of connected shafts intersect at the center point of the coupling, but not in the same straight line (Figure 2.3B).
3. Combined angular and offset. The axes of connected shafts do not intersect at the center point of the coupling and are not parallel (Figure 2.3C).

It is important to recognize that while the equipment may see these

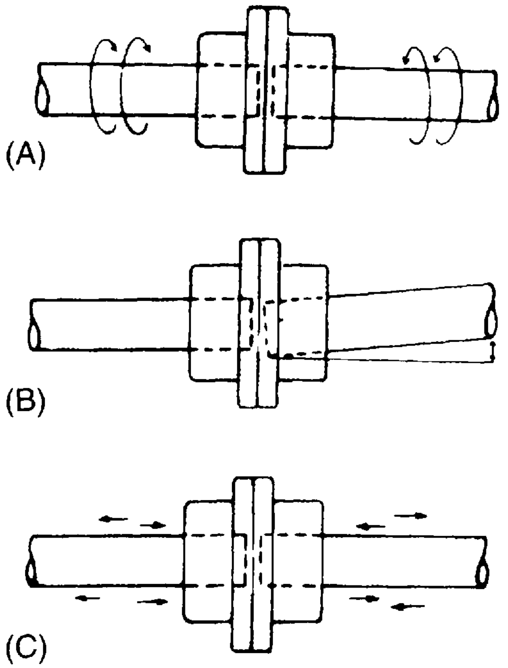


Figure 2.2 Functions of a flexible coupling: (A) Transmit torque; (B) accommodate misalignment; (C) compensate for end movement.

types of misalignment, the coupling sees only angular misalignment. The *flexible elements* see only angular and axial alignment or misalignment. Therefore equipment needs a coupling with more than one flexible element to accommodate offset, the exception being some elastomeric element couplings that can accommodate both parallel and angular misalignments.

C. Accommodate End Movement

Most flexible couplings are designed to accommodate axial movement of the equipment to shaft ends. In pumps that are driven by motors, the couplings are usually required to limit the axial float of the equipment

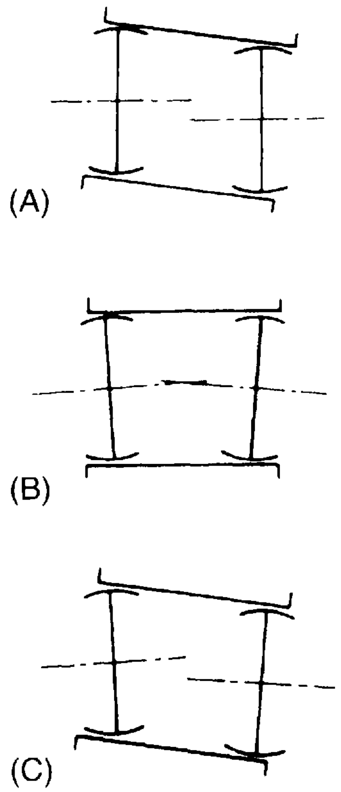


Figure 2.3 Types of misalignment. (A) Parallel offset misalignment. (B) Angular misalignment. (C) Combined angular and offset misalignment.

to prevent internal rubbing in the motor. Some pumps are driven by motors with sleeve bearings that do not have thrust bearings.

Misalignment and end movement must be accommodated without the introduction of abnormal loads in the equipment. Generally, machines are set up at installation quite accurately. Many things can force equipment to run out of alignment. The thermal effects of handling hot and cold fluids cause some movement in the vertical and axial directions, as do differentials of temperature in driver media such as gas and steam. Vertical motion could be a result of support structure expansion due to temperature differences, distortion due to solar heating, axial growth, or a combination of any of these. Horizontal motion is usually due to piping forces caused by poor installation practices and expansion or contraction caused by changes in the temperature or pressure of the media in the system.

It is a fact of life that machinery appears to live and breathe, move, grow, and change form and position; this is one of the basic reasons for using flexible couplings. However, a flexible coupling is not the solution to all movement problems that can exist in a sloppy system. It is naive to use a flexible coupling in the hope that it will compensate for any and all motions. Flexible couplings have their limitations. The equipment or system designer must make calculations that will give a reasonable estimate of the outer boundaries of the anticipated gyrations. Unless those boundaries are defined, the equipment or system designer may just be transforming equipment failure into coupling failure (see Figure 2.4).

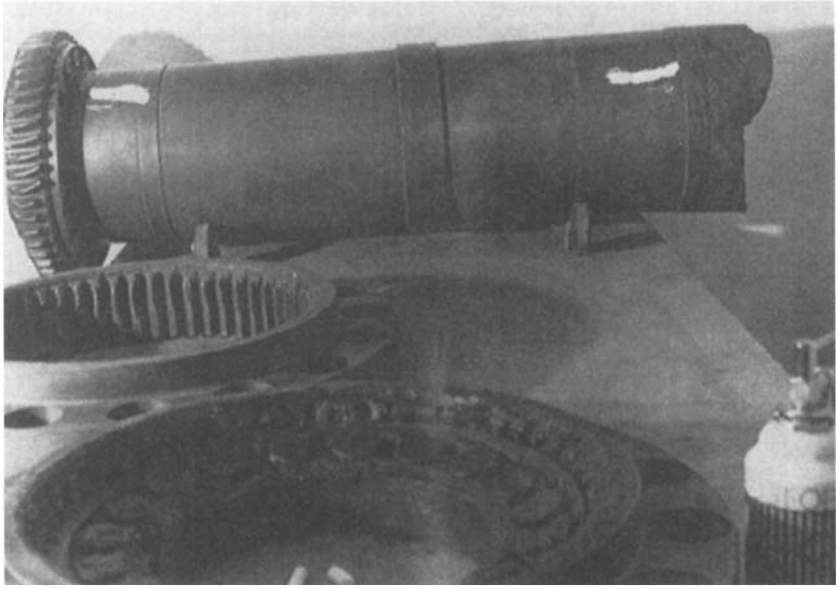


FIGURE 2.4 Coupling failure.

One thing to remember is that when subjected to torque and misalignment, *all* couplings react on the connected equipment components. Some produce greater reactionary forces than others, and if these forces are overlooked they can cause vibration, shaft failure, bearing failure, and/or other operational and early failure of other components of the drivetrain (see Figure 2.5). It is important for the equipment or system designer not to confuse the term "coupling misalignment capacity" with "equipment misalignment tolerance." The capabilities of a coupling are usually substantially higher than the equipment can accept (see Figure 4.7 for coupling misalignment limits and Section III of Chapter 4 for guidelines for equipment misalignment tolerances).

Rigid couplings produce the greatest reactions on equipment. Mechanically flexible types such as gear, chain, and grid couplings produce

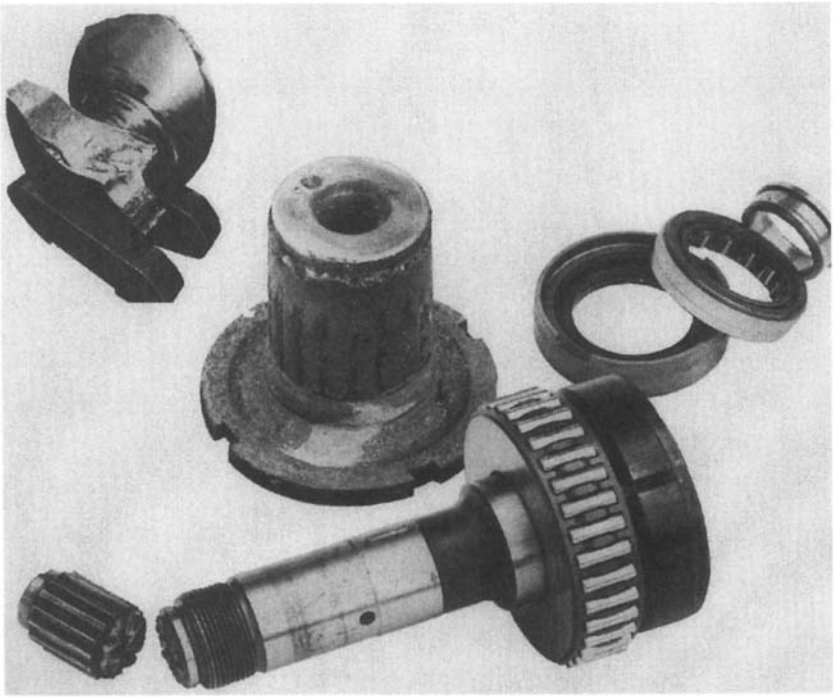


FIGURE 2.5 Equipment failure.

moderate to high moments and forces on equipment that are a function of torque and misalignment. Elastomeric element couplings produce moderate to low moments and forces that are slightly dependent on torque. Metallic element couplings produce relatively low moments and forces that are relatively independent of torque. The most commonly used flexible couplings today are those that exhibit the greatest flexibility (misalignment and axial capacity) while producing the lowest external loads on the coupled equipment.

D. Other Functions of Flexible Couplings

Besides their basic functions, flexible couplings are sometimes required to

1. Dampen vibration and reduce peak or shock loads
2. Protect equipment from overload
3. Measure the output torque of driven equipment
4. Insulate the driver from the driven equipment
5. Position a rotor of a motor or generator
6. Be used to tune a system out of a torsional critical mode

II. TYPES OF FLEXIBLE COUPLINGS—GENERAL

There are many types of couplings (see Figure 2.6). Virtually all of them can be put into two classes, rigid or flexible. Flexible couplings can be divided into four categories—mechanically flexible, elastomeric, metallic element, and miscellaneous—and three industrial disciplines—miniature, general-purpose, and special-purpose.

A. Types of Couplings

The two classes of couplings are (1) rigid couplings—(a) rigid flanged coupling (Figure 2.7), (b) rigid ribbed coupling (Figure 2.8), (c) rigid sleeve coupling (Figure 2.9), or (d) quill shaft coupling (Figure 2.10); and (2) flexible couplings.

Rigid coupling are discussed in Chapter 5.

There are four basic types of flexible couplings:

1. Mechanically flexible couplings (mechanical element or contact couplings)

Rigid Couplings	***Flexible Couplings***	***Flexible Couplings***	***Flexible Couplings***
	Miniature Flexible Couplings	**General Purpose Industrial Couplings**	**Special Purpose Industrial Couplings**
Flanged Rigid Coupling	**Mechanically Flexible** **Gear Coupling** Non-Lubricated Gear Coupling	**Mechanically Flexible** **Gear Coupling** Straight Tooth Gear Coupling Crowned Tooth Gear Coupling **Chain Coupling** Chain Coupling- Steel Chain Coupling - Nylon **Grid Coupling** Grid Coupling- Vertical Split Cover Grid Coupling- Horizontal Split Cover	**Mechanically Flexible** **Gear Coupling** High Angle (Gear Spindle) High Speed (sealed lubed) High Speed (continuously lubed) High Speed (continuously lubed marine style)
Rigid Ribbed Coupling	**Elastomeric Element** Urethane Ribbon Elastomeric Coupling	**Elastomeric Element** **Shear Type** Urethane Tire Coupling Corded Tire Coupling Elastomeric Donut Coupling (Unclamped) **Compression Type** Compression Donut Coupling Block Coupling Jaw Coupling Pin and Bushing Coupling	**Elastomeric Element** **Shear Type** Bonded Shear Coupling **Compression Type** Block Coupling
Rigid Sleeve Coupling	***Metallic Element*** **Metallic Beam Coupling** **Disc Coupling** **Metallic Bellows Coupling**	***Metallic Element*** **Disc** Circular Disc Coupling Square Disc Coupling Scalloped Disc Coupling Link Disc Coupling	***Metallic Element*** **Disc** Reduced Moment Disc Coupling (scalloped) Marine Style Disc Coupling (scalloped) Frame Disc Coupling **Diaphragm** Taper Diaphragm Coupling (Welded) One Piece Diaphragm Coupling Multiple Straight Diaphragm Coupling Multiple Convoluted Diaphragm Coupling
Quill Shaft Coupling		**Miscellaneous** Pin and Bushing Coupling Metallic Beam Coupling Sliding Block Coupling	**Miscellaneous** Offset Coupling (Schmidt Coupling) Tangential Spring Coupling

FIGURE 2.6 Types of couplings.

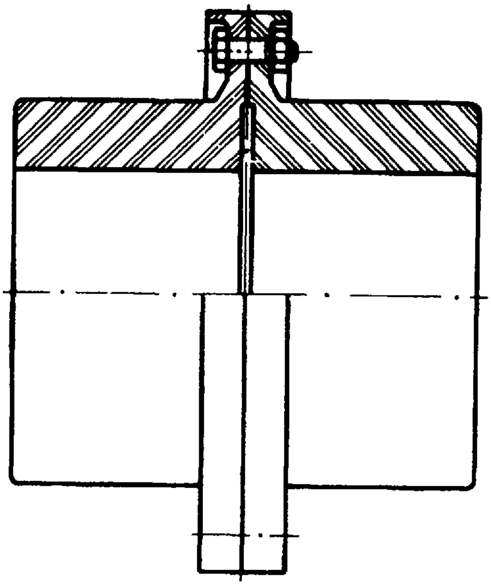


Figure 2.7 Rigid flanged coupling.

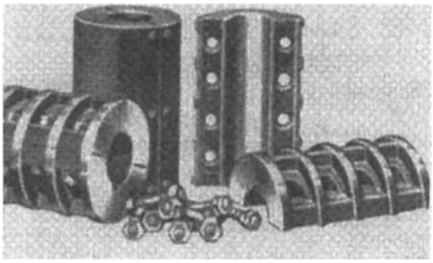


Figure 2.8 Rigid ribbed coupling (compression coupling).

2. Elastomeric element couplings
3. Metallic element couplings
4. Miscellaneous couplings

The mechanically flexible types generally obtain their flexibility from loose-fitting parts from the rolling or sliding of mating parts, or from both. They require lubrication unless one moving part is made of

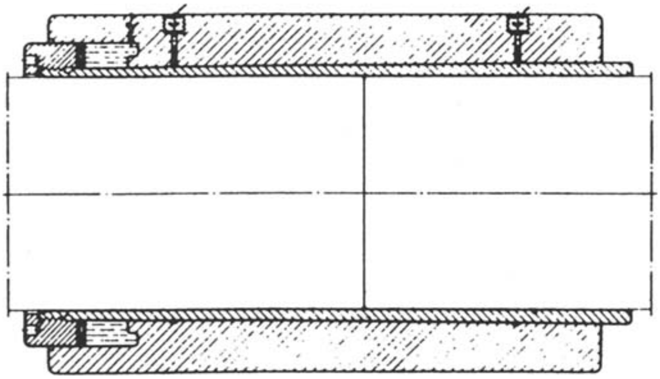


FIGURE 2.9 Rigid sleeve coupling. (Courtesy of SKF Steel, Coupling Division.)

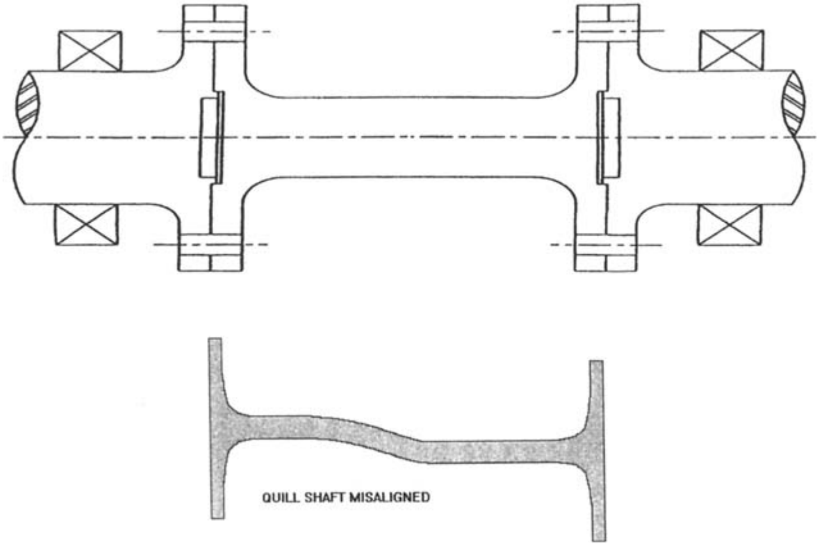


FIGURE 2.10 Quill shaft coupling.

a material that meets its own lubrication need (i.e., a nylon gear coupling).

The elastomeric element types obtain their flexibility from stretching or compression of a resilient material (rubber, plastic, etc.).

The metallic element types obtain their flexibility from the flexing of thin metallic discs or diaphragms.

Miscellaneous coupling types obtain their flexibility from a combination of the mechanisms just described or through a unique mechanism.

B. The Disciplines for Flexible Coupling Applications

There are three disciplines for the application of flexible couplings:

1. The miniature discipline, which covers couplings used for office machines, servomechanisms, instrumentation, light machinery, and so on:
   a. Miniature nonlubricated gear coupling (Figure 2.11)
   b. Miniature urethane ribbon elastomeric coupling (Figure 2.12)

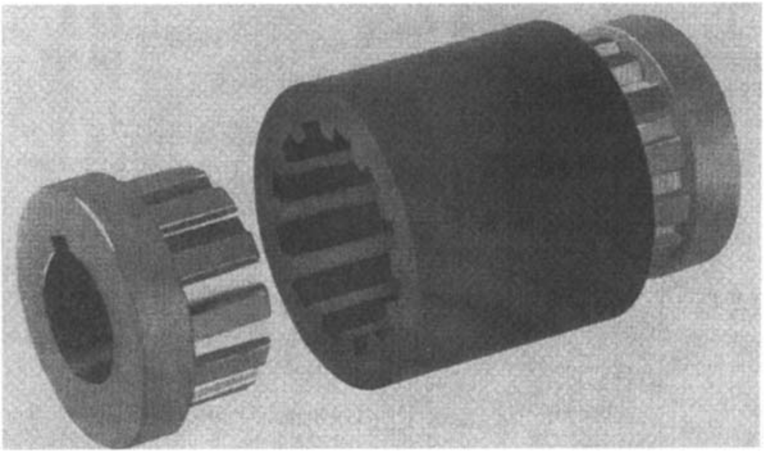


Figure 2.11 Miniature nonlubricated gear coupling. (Courtesy of Guardian Industries, Inc.)

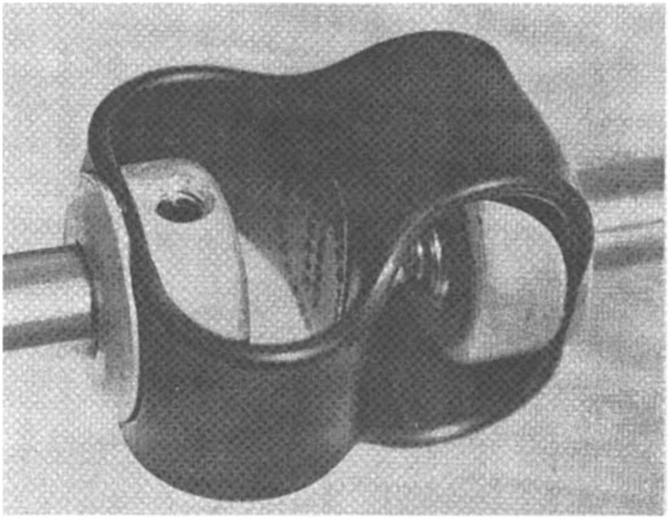


Figure 2.12 Miniature urethane ribbon elastomeric coupling. (Courtesy of Acushnet Company.)

   c. Miniature metallic beam coupling (Figure 2.13)
   d. Miniature disc coupling (Figure 2.14)
   e. Miniature metallic bellows coupling (Figure 2.15)

2. The general-purpose industrial discipline, which covers couplings used in the steel industry, the petrochemical industry, utilities, machinery, and so on. Most of these couplings are considered "off the shelf" or modifications of them.
3. The special-purpose industrial discipline, which covers couplings that are designed for a specific application or piece of equipment and are generally not "off the shelf."

III. TYPES OF GENERAL-PURPOSE COUPLINGS

General-Purpose couplings can be classified according to "size." Couplings for operations under 100 hp are classified as small. Those between 100 and 1000 hp are classified as medium: Usually those over 1000 hp are considered critical to the operation of the equipment: these are covered in Section IV, Special-Purpose Couplings. If a small piece

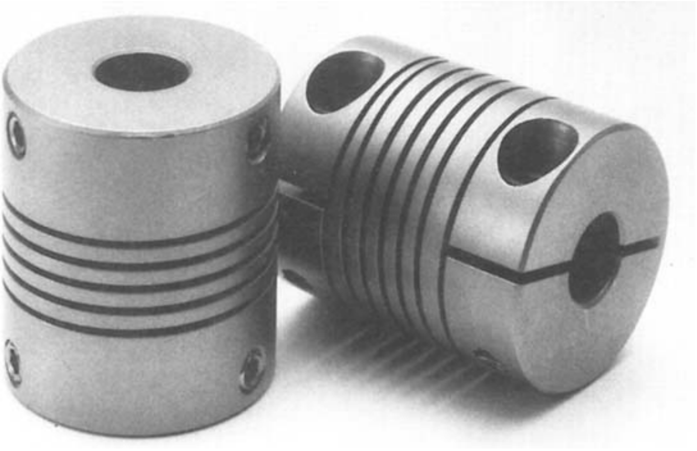


Figure 2.13 Miniature metallic beam coupling. (Courtesy of Helical Products Company, Inc.)

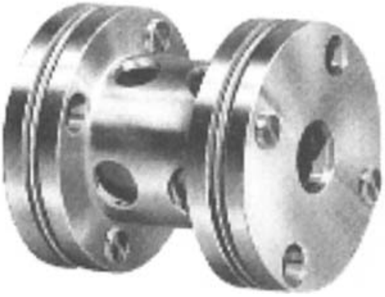


Figure 2.14 Miniature disc coupling. (Courtesy of Coupling Division of Rexnord, Thomas Couplings.)

of equipment (e.g., a pump) shuts down, it usually does not affect the plant operation. This equipment uses a coupling with a flexible element that can be easily inspected and replaced and is therefore often considered a throwaway. The couplings are very flexible and require very simple alignment techniques, calipers, scales, and perhaps, if one gets sophisticated, a dial indicator.

A failure from overtorque or overmisalignment is usually a failure

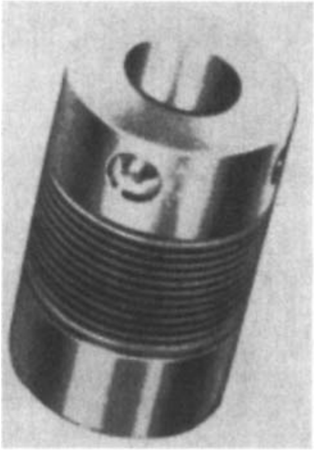


FIGURE 2.15 Miniature metallic bellows coupling. (Courtesy of Metal Bellows Corporation.)

of the flexible element, and usually little or no damage occurs to other components. A few examples of these types of couplings on the market are grid, disc, and elastomeric. Also found on this type of equipment are small gear couplings, some of which have the outer sleeve made of nylon or plastic and therefore require no lubrication.

Medium-sized equipment (100–1000 hp) is not normally critical to the operation of the plant but can be problematic and costly if constant maintenance and downtime are required. It uses grid, gear, disc, and elastomeric couplings. The decision as to which type of coupling to use is probably based on what comes with the equipment. Each equipment manufacturer bases its selection on past price, experience, and a preference to work with one coupling manufacturer rather than another. Usually if a coupling fails, the customer's plea is "Get me one just like the one that failed." But if many problems occur—failures, too much time to install, the need for constant lubrication, and numerous maintenance problems—someone starts to question whether there is something better "out there." Over time, maintenance departments eventually develop a preference based on pump duty, type of maintenance performed, and, sometimes, just their own personal preference. This group of couplings usually fail because of improper installation, inadequate lubrication, or improper alignment.

A. Mechanically Flexible Couplings

1. Gear Couplings

Gear couplings consist of two hubs with external teeth that engage internal teeth on a two-or one-piece sleeve. The teeth may be straight (see Figure 2.16) or curved (crowned) (see Figure 2.17). For applications requiring over 1/2° of Flexibility, curved teeth may be better. These couplings obtain their flexibility from the looseness (backlash) between the mating teeth. Gear couplings are used for medium-sized and large equipment applications and are probably the most power-dense type of couplings available. They require periodic lubrication, every 1–2 years, depending on the work duty and type of lubrication. If properly main-

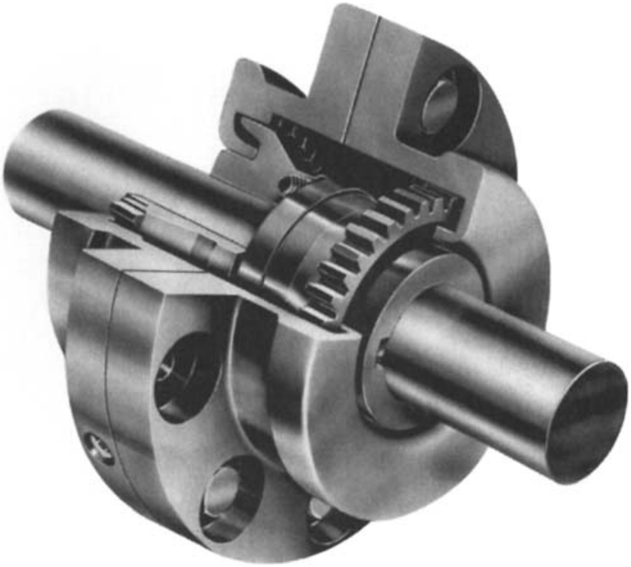


FIGURE 2.16 Straight tooth gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

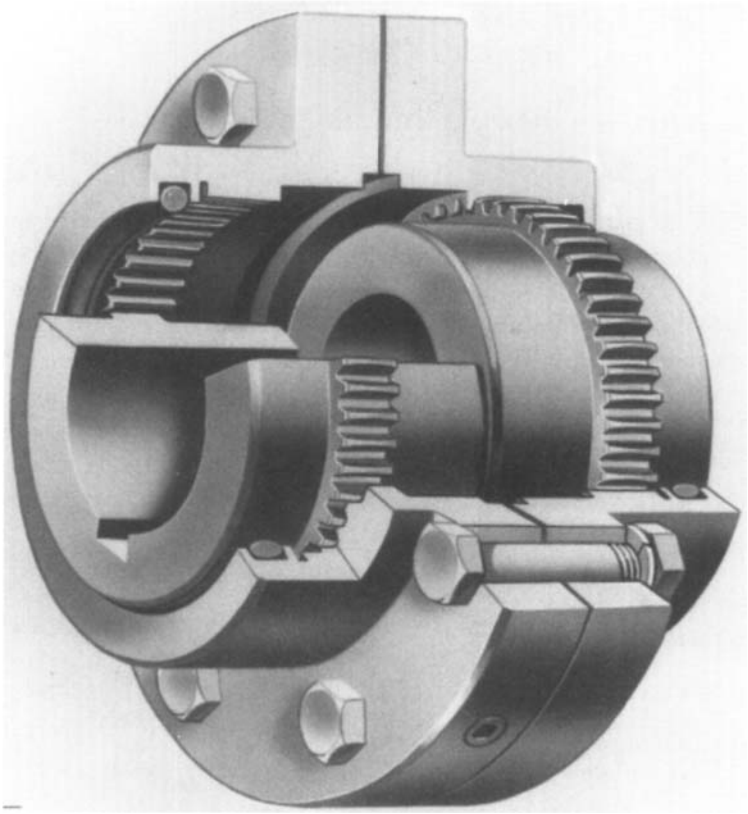


FIGURE 2.17 Crowned tooth gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

tained (good lubrication and reasonable alignment), gear couplings have a service life of 3–5 years, in many cases 5–10 years.

Some gear couplings have a sleeve that is made of plastic (nylon, high molecular weight plastic); these do not require lubrication. They have much lower torque capacity than all-steel couplings and are used mainly on small pumps.

2. Chain Couplings

Chain-type couplings (see Figure 2.18) are very similar to gear couplings. They are usually composed completely of metal (usually the

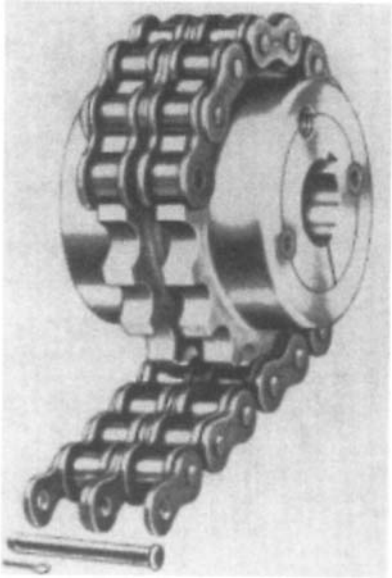


FIGURE 2.18 Chain coupling—steel. (Courtesy of Dodge Division of Reliance Electric.)

sprockets and chains are hardened), but some have plastic chains (see Figure 2.19). They have two hubs with sprockets rather than teeth. The sprockets are connected by a length of double-stranded roller chain. Most of the couplings are open and are lubricated by "brushing" grease on the chain. Sometimes a cover is used to help keep the lubrication in. These couplings do not transmit as much power (for the same outside diameter) as gear couplings but are usually less costly. Chain couplings are used for low-speed applications on medium-sized and small equipment.

3. Grid Couplings

Grid couplings are also very similar to gear couplings. Usually composed of all metal, they have some degree of resilience. These couplings can dampen vibration and reduce peak or shock loads by 10–30%. They have two hubs with serrations (grooves) rather than teeth. The grooves are connected by a steel grid. A cover keeps the lubrication in. The covers are either vertically split (see Figure 2.20) or horizontally split (see

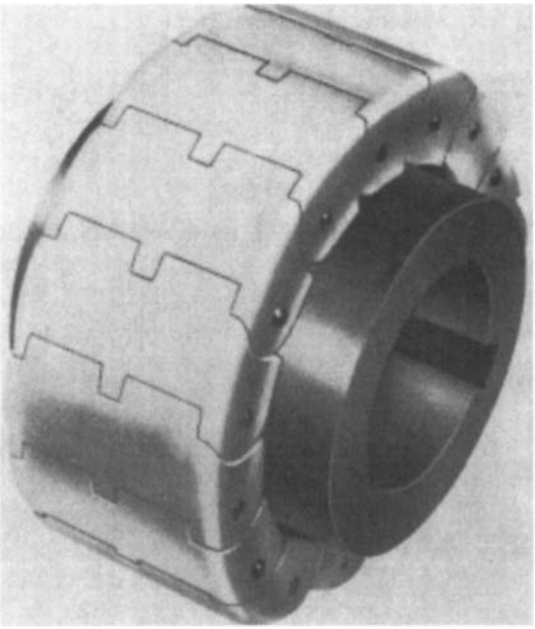


FIGURE 2.19 Chain coupling—nylon. (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

Figure 2.21). Grid couplings do not transmit as much power (for per the same outside diameter) as gear couplings but are usually less costly. Grid couplings are used for medium-sized and small equipment.

4. U-Joints

U-joints are used in many general-purpose applications. They are used primarily on vehicles (see Figure 2.22). There are several types used here. Rear wheel drives use the "pin-and-yoke" type. Steering wheels use U-joints (see Figure 2.23) to accommodate large misalignments; these do not rotate, but they do transmit power and accommodate misalignment. In industrial applications, the yoke-type U-joints (see Figure 2.24) are used in the paper and steel industries.

U-joints are generally used in applications where a high degree of misalignment must be accommodated (generally in excess of 1 degree).

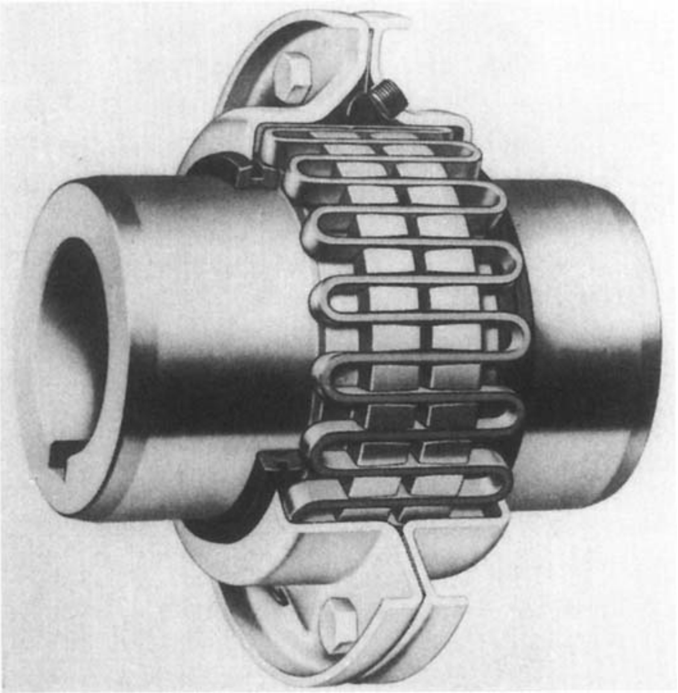


FIGURE 2.20 Grid coupling—vertical split cover. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

They can accommodate end movement by using a telescoping shaft (square shafting or splines). U-joints function by means of a sliding motion between two flanges shaped like a fork (a yoke) and having a hole (eye) radially through the yoke that is connected by a cross. This allows flexibility by sliding; a more sophisticated design puts bearings and seals between the yoke and the cross (most industrial U-joints use bearings and seals), giving us the modern-day U-joint.

B. Elastomeric Element Couplings

1. Urethane Tire Couplings

Common on small equipment, urethane tire couplings (see Figure 2.25) have their elastomeric element in shear and are made of urethane. The tires are split to enable easy assembly without removing the hubs. Ure-

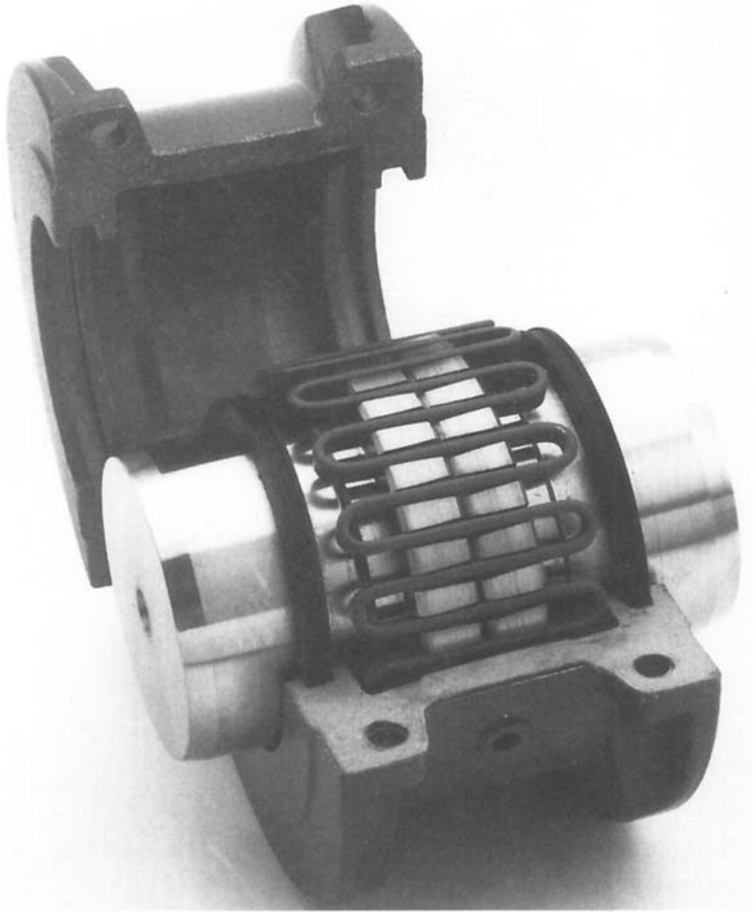


Figure 2.21 Grid coupling—horizontal split cover. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

thane couplings offer a high degree of flexibility. When the coupling fails, usually only the elastomeric element is replaced.

2. Corded Tire Couplings

Corded tire couplings (see Figure 2.26) also have their elastomeric elements in shear. They use a reinforced element (similar to belted auto tires). Because of the reinforcement in the element the torque capacity

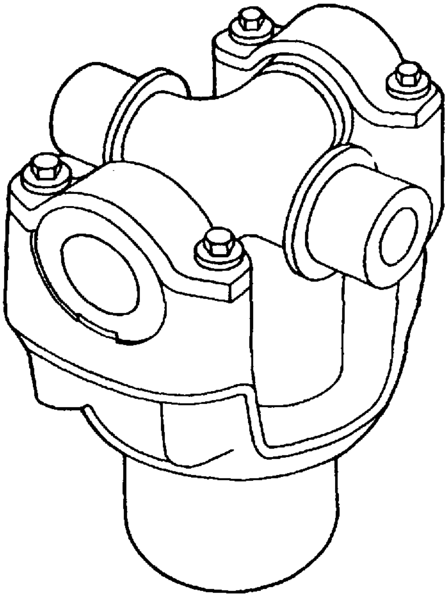


FIGURE 2.22 Vehicle U-joint.

is greater per unit outside diameter than that of the urethane tire type. Most small and some medium-sized equipment applications use corded tire couplings.

3. Elastomeric Donut Couplings—Unclamped

Another type of coupling common on small equipment, elastomeric donut couplings (see Figure 2.27) have their elastomeric elements in shear. They provide low torsional stiffness and low reactionary forces. Oversizing can lead to premature failure by wearing of the engaging teeth rather than material failure flexing.

4. Compression Donut Couplings

Compression donut couplings (see Figure 2.28) have a precompressed elastic element. Screws force the donut to a smaller diameter. All legs of the donut are in compression before the load is applied. Medium-sized and some small equipment incorporate these couplings.

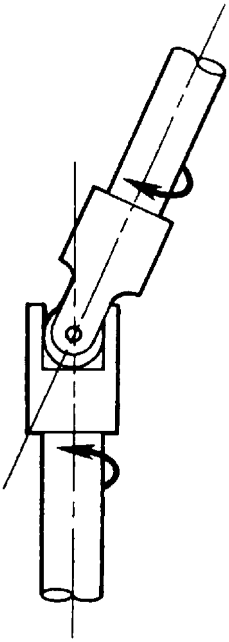


FIGURE 2.23 Mechanism U-joint.

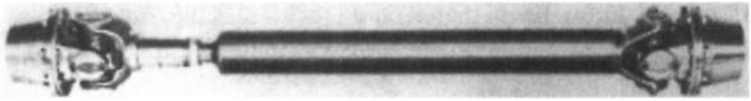


FIGURE 2.24 Yoke-type U-joint. (Courtesy of Spicer Universal Joint, Division of Dana Corporation.)

5. Block Couplings

Block couplings (see Figure 2.29) use rubber in compression. The rubber blocks are installed in cavities formed by internal sleeve blades, an external hub blade, and two end plates. This type is unique among couplings due to its fail-safe feature. If the elastomeric element fails, the coupling may run for some time on the metal blades. These couplings can provide up to 1/4 degree of misalignment and parallel offset capabilities of 1/64–1/32 in. Sometimes medium-sized and small equipment

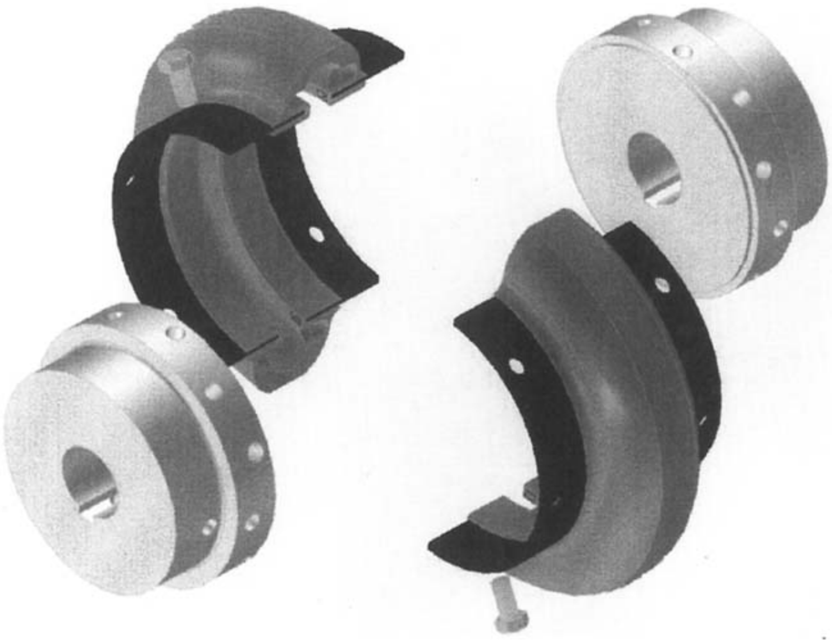


FIGURE 2.25 Urethane tire coupling. (Courtesy of Rexnord, Thomas Couplings.)

will incorporate these couplings because if properly aligned they require no maintenance (except for replacement of blocks every 3–5 years).

6. Jaw Couplings

Jaw couplings (see Figure 2.30) have their elastomeric elements in compression. The flexible element can be one piece or spilt to facilitate replacement. These couplings also have a fail-safe feature. Flexible elements are made of many types of elastomeric materials such as rubber or urethane. The properties (hardness, resiliency, etc.) can be varied to suit the required loads. These couplings are used primarily to accommodate misalignment and transmit power. Small and medium-sized equipment employs these couplings. Be careful, since sometimes they are used to absorb energy and dampen loads, particularly when equip-

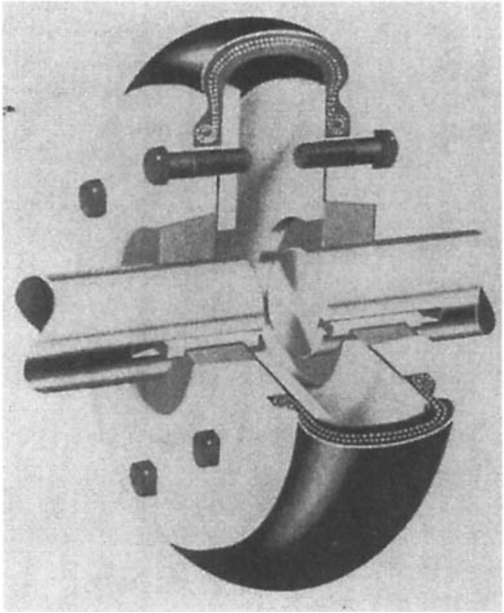


Figure 2.26 Corded tire coupling. (Courtesy of Dayco Corporation.)

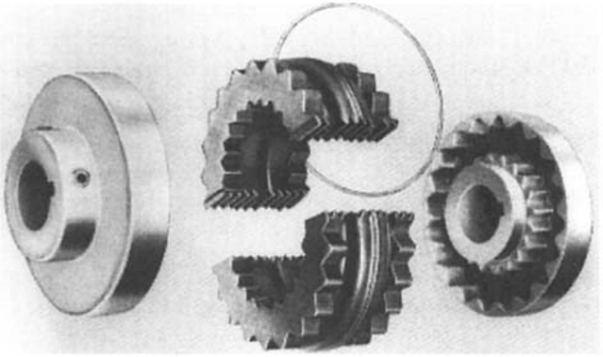


Figure 2.27 Elastomeric donut coupling (unclamped). (Courtesy of TB Wood's Sons Company.)

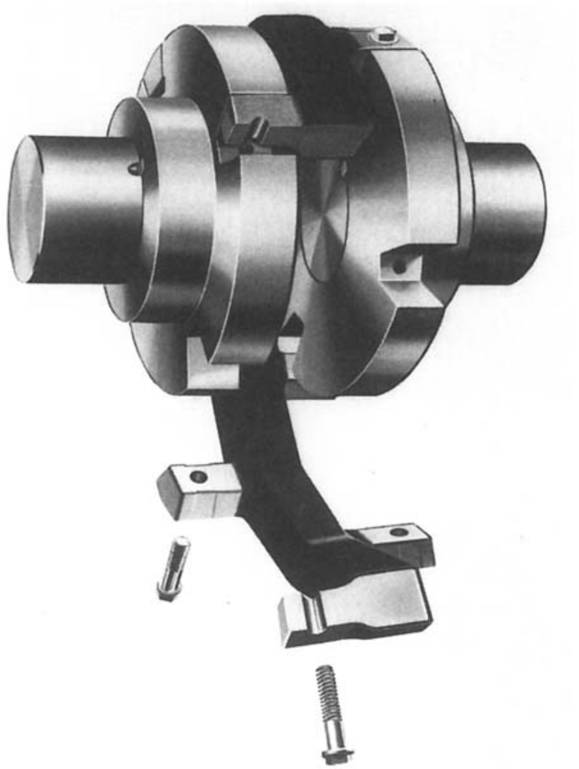


FIGURE 2.28 Compression donut coupling. (Courtesy of Kop-Flex, Inc. Emerson Power Transmission Corporation.)

ment is engine-driven. Also, because they are torsionally soft, they are sometimes used to tune a system out of a torsional critical mode.

7. Pin-and-bushing Couplings

Pin-and-bushing couplings (see Figure 2.31) are used primarily to accommodate misalignment and transmit power. Relative movement is accommodated by compression of steel bonded bushings. Small and medium-sized equipment employs these couplings. They are used to absorb energy and dampen loads, particularly when the equipment is engine-driven.

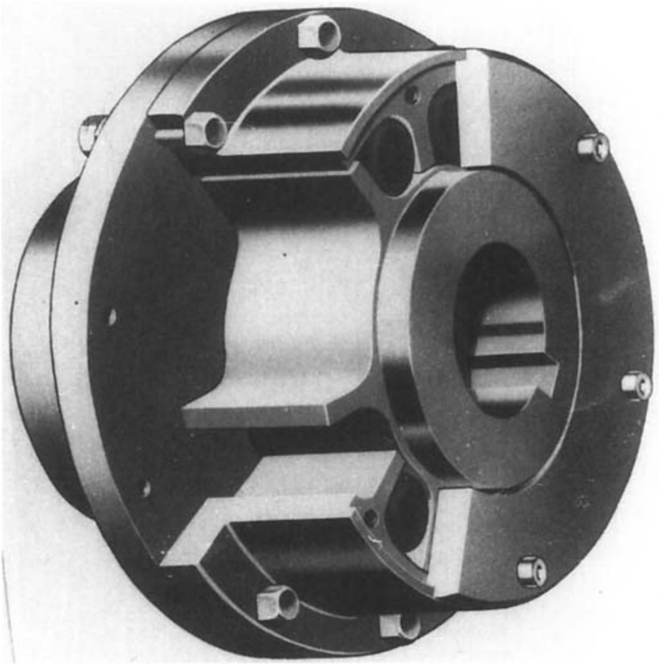


FIGURE 2.29 Block coupling. (Courtesy of Kop-Flex, Inc. Emerson Power Transmission Corporation.)

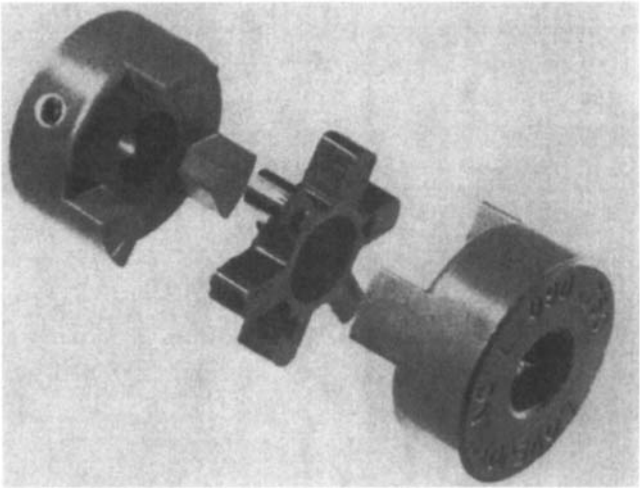


FIGURE 2.30 Jaw coupling. (Courtesy of Lovejoy, Inc.)

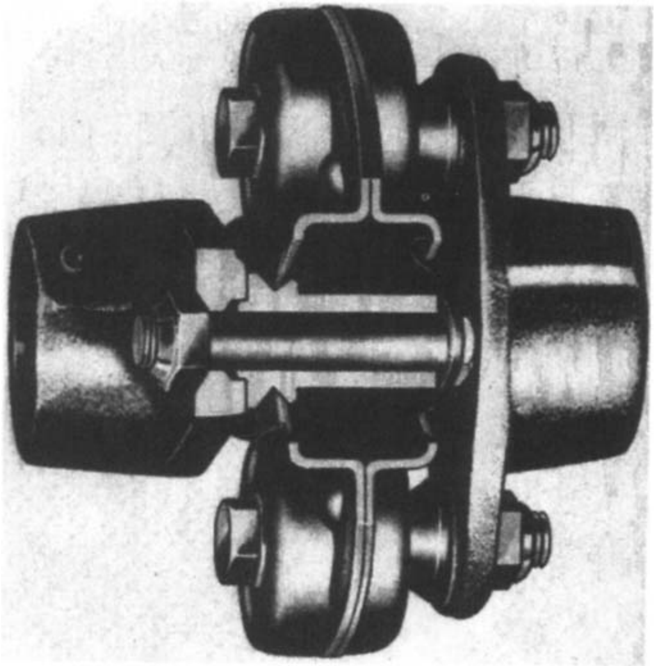


Figure 2.31 Pin-and-bushing coupling. (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

C. Metallic Element Couplings (Disc Couplings)

Disc couplings transmit torque by a simple tensile force between alternating driving and driven bolts on a common bolt circle. Misalignment arises from the flexibility that comes from the length of material between the bolts. Disc couplings have been around for years, but with the use of finite element analysis, they can be and have been optimized for optimum characteristics. These couplings are composed completely of metal and do not require lubrication. The discs are usually continuous and of various shapes—circular (see Figure 2.32), square (see Figure 2.33), or scalloped (see Figure 2.34)—but can also be individual links (see Figure 2.35). Most disc couplings use multiple thin discs rather than one thick disc or link because stresses from misalignment are a function of T^3 vs. Nt^3. These couplings are used in medium-sized equip-

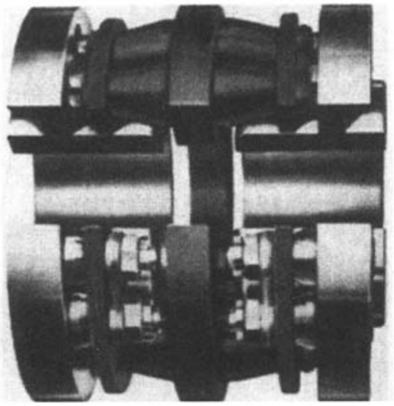


FIGURE 2.32 Circular disc coupling. (Courtesy of Rexnord, Thomas Couplings.)

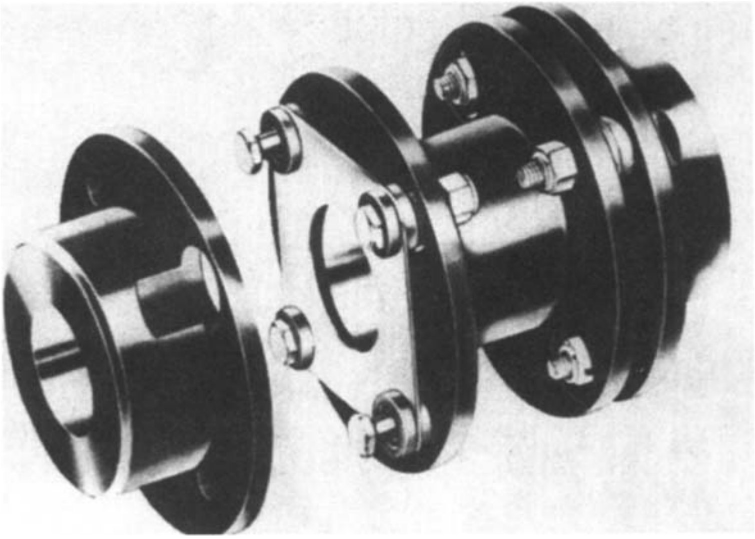


FIGURE 2.33 Square disc coupling. (Courtesy of TB Wood's Inc.)

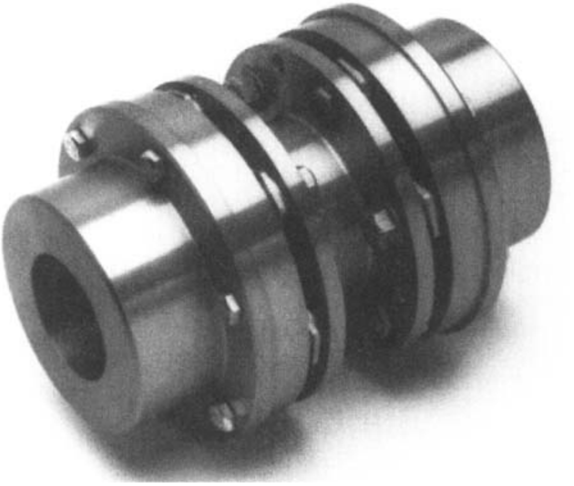


Figure 2.34 Scalloped disc coupling. (Courtesy of Kop-Flex, Inc. Emerson Power Transmission Corporation.)

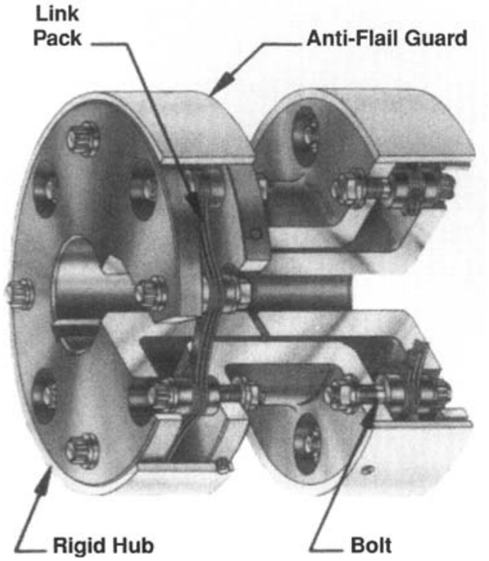


Figure 2.35 Link disc coupling. (Courtesy of Ameridrives International.)

ment applications. If the misalignment is beyond 1/2° during operation, the flexible element may fail in fatigue.

D. Miscellaneous Couplings

1. Metallic Beam Couplings

Metallic beam couplings (see Figure 2.36) transmit torque through a spiral-cut beam. The beams are shaped to allow torque to be transmitted in either direction. Misalignment is accommodated by deflection of the curved beam. Metallic beam couplings can accommodate misalignment up to 1/2°. These couplings are used in small equipment applications.

2. Sliding Block Couplings

Sliding block couplings (see Figure 2.37) transmit torque through an intermediate square floating member and compensate for all three types of misalignments by sliding action between the closely fitted center member and the adjacent driving and driven jaw flanges. These couplings are used in small and sometimes medium-sized equipment applications. *Note:* Some forms of this type of coupling are known as Oldham couplings.

IV. SPECIAL-PURPOSE COUPLINGS

There are two categories of special-purpose couplings: lubricated and nonlubricated. There is a lot of older equipment that uses lubricated couplings still around whereas most new equipment or upgrade equipment uses low maintenance couplings that require infrequent lubrica-

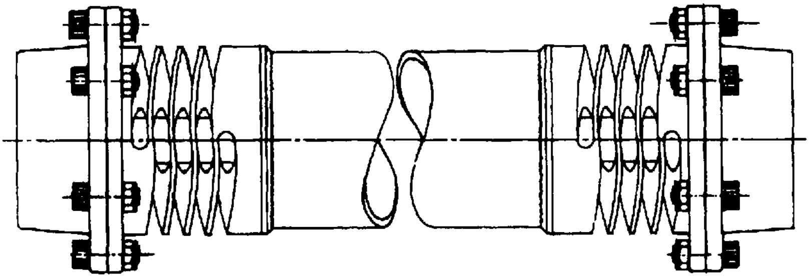


FIGURE 2.36 Metallic beam coupling. (Courtesy of Panamech Company.)

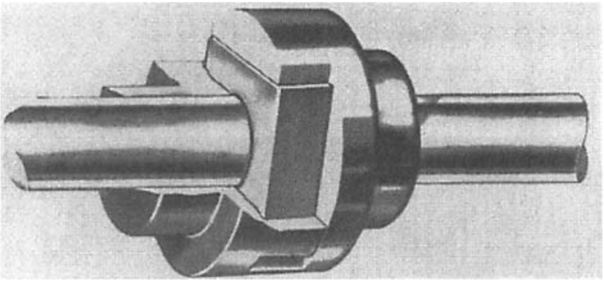


Figure 2.37 Sliding block coupling. (Courtesy of Ameridrives International.)

tion; the tendency is to use nonlubricated couplings. Generally these are gear couplings (mechanically flexible couplings) that require oil or grease lubrication and flexible element couplings that require no lubrication. The gear coupling category has many subgroups to allow for the many different designs; however, they are only variations.

Gear couplings are usually made of alloy steel and have surface hardened teeth (usually nitrided). U-joints also require lubrication, but they have positive seals that do not flex under misalignment. There are two primary types of nonlubricated couplings (metallic element couplings) that are used for special-purpose applications: the diaphragm type and the disc type.

A. Mechanically Flexible Couplings

1. High-angle Gear Couplings

The high-angle gear couplings (gear spindles) (see Figure 2.38) are used on heavy duty, high-torque applications such as metal rolling mill main drives. The gear teeth of these spindles are usually made of alloy steels and are surface hardened (induction hardened, nitrided, or carburized). Which material and heat treatment is best depends on the specific application.

2. High-performance Gear Couplings

The sealed lubricated gear coupling (see Figure 2.39) was adapted and modified during the 1950s and 1960s to carry more power at higher speeds. The major problem with gear couplings is lubrication. Grease

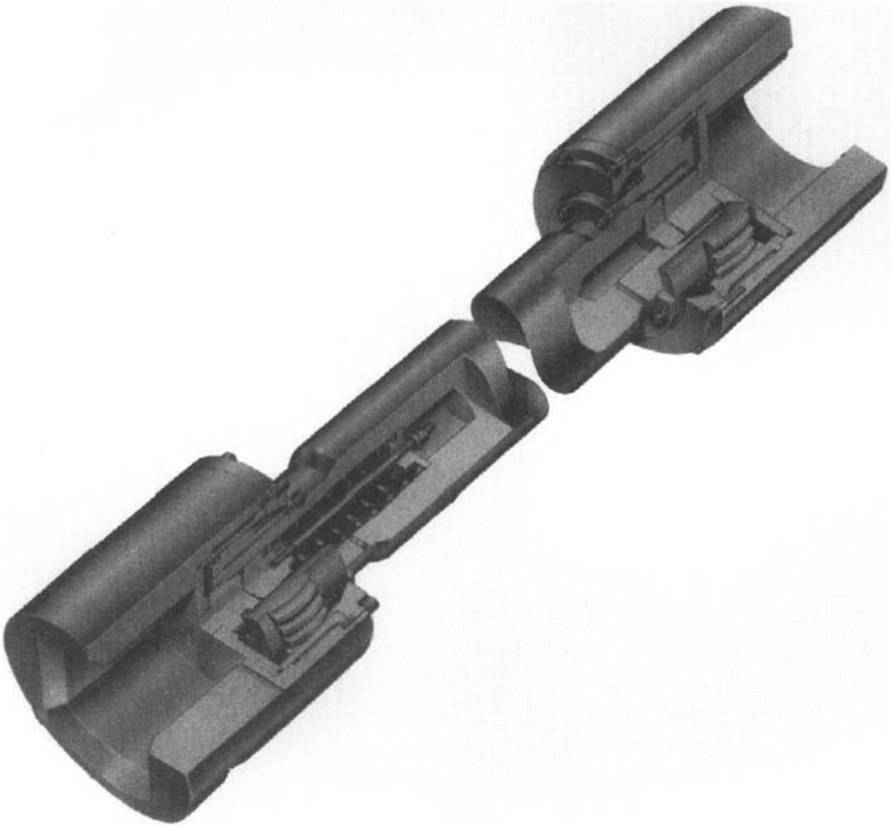


FIGURE 2.38 High-angle gear coupling (gear spindle). (Courtesy of Renolds, Inc., Engineered Products Division.)

separates under centrifugal force, and seals leak. Continuously lubricated designs (see Figure 2.40) that use the machines bearing oil are more common for this type of application. There are several design variations, couplings with external teeth on the spacer in the "marine" type (see Figure 2.41). This design can be used with integral flanges on the machine shafts such as on ship propulsion turbines and gears—hence the term "marine type." Reduced moment versions were readily also used. If proper lubrication is maintained, these couplings will operate

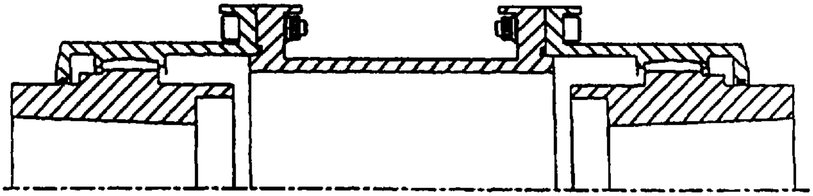


Figure 2.39 High-speed gear coupling (sealed lubed).

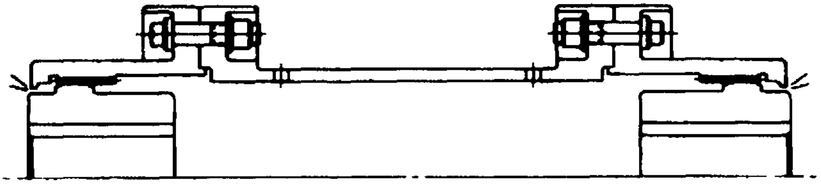


Figure 2.40 High-speed gear coupling (continuously lubed).

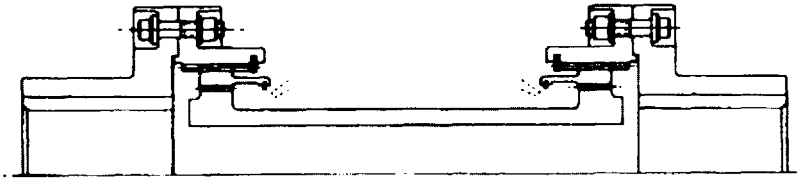


Figure 2.41 High-speed gear coupling (continuously lubed, marine style).

successfully for years. Lubrication or lack of it is the biggest cause of failure. The coupling itself tends to act as a centrifuge and separates particles out of the oil. This produces "sludge" (see Figure 2.42), which causes the coupling to lock up, which in turn causes the equipment to vibrate. The most common mode of failure for a gear coupling is wear due to lubrication problems.

3. U-joints

U-joints (or driveshafts) are used in many special-purpose applications. There are several types used for various applications. Front wheel

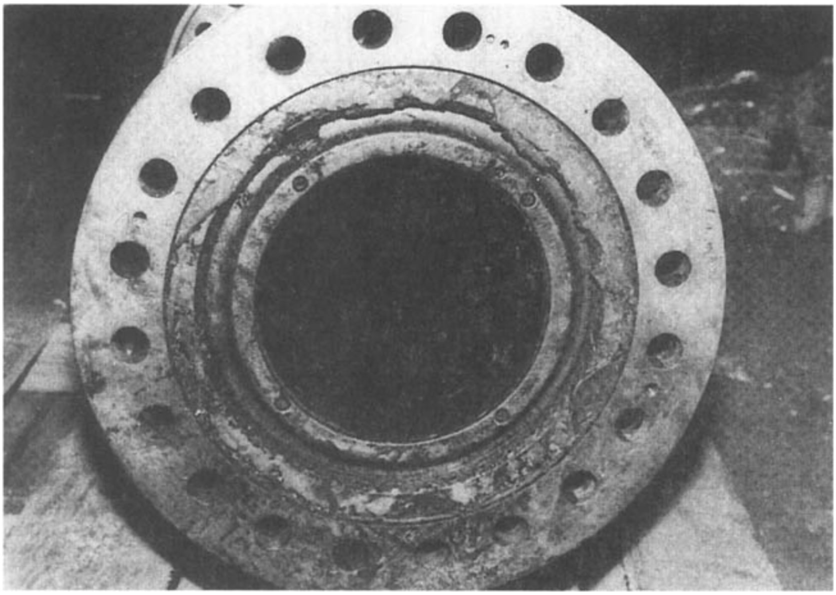


FIGURE 2.42 Sludged gear coupling.

drives of vehicles use constant-velocity joints (see Figure 2.43). Constant-velocity U-joints are also used on test stands for many types of rotating equipment (engines, compressors, etc.) because the equipment can easily be installed without much concern about how accurately it is aligned. Most special U-joints that are used in industrial applications are used in the paper and steel industries.

U-joints are generally used in applications where high misalignment must be accommodated (generally in excess of 1°). There are two basic types of U-joints used: the yoke type (see Figure 2.44) and the block type (see Figure 2.45). They can accommodate end movement by using a telescoping shaft (square shafting or splines) or, on large universals for steel mills where the misalignment changes with the reduction of material and therefore the movement is small, they compensate for end float by sliding on the roll bore. Many of today's U-joints have been optimized by FEA, are made of modern materials, and have been heat-treated; therefore they have substantially high torque ratings at

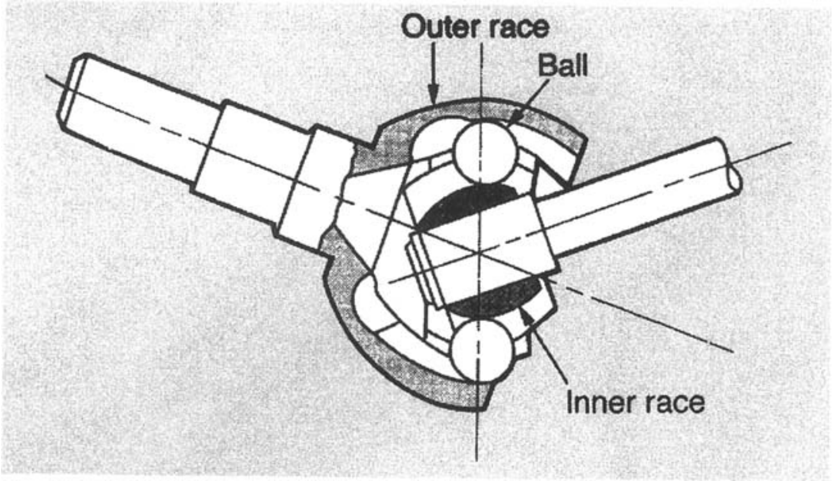


FIGURE 2.43 Constant-velocity U-joint.

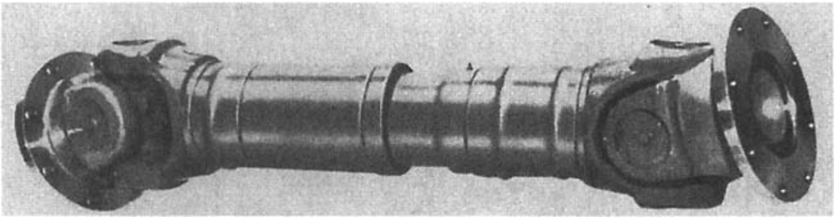


FIGURE 2.44 Yoke-type U-joint. (Courtesy of Voithe Transmit GmbH.)

misalignments. At 1° of misalignment a gear spindle has higher torque capacity than a U-joint (see Figure 2.46). At 3° of misalignment, a U-joint will meet (or exceed) the capacity of a gear spindle. At over 3° of misalignment, a U-joint usually has higher torque capacity than a gear spindle.

B. Elastomeric Element Couplings

1. Bonded Shear Couplings

Bonded shear couplings (see Figure 2.47) are used in special applications that require specific torsional stiffness to tune the system (such as

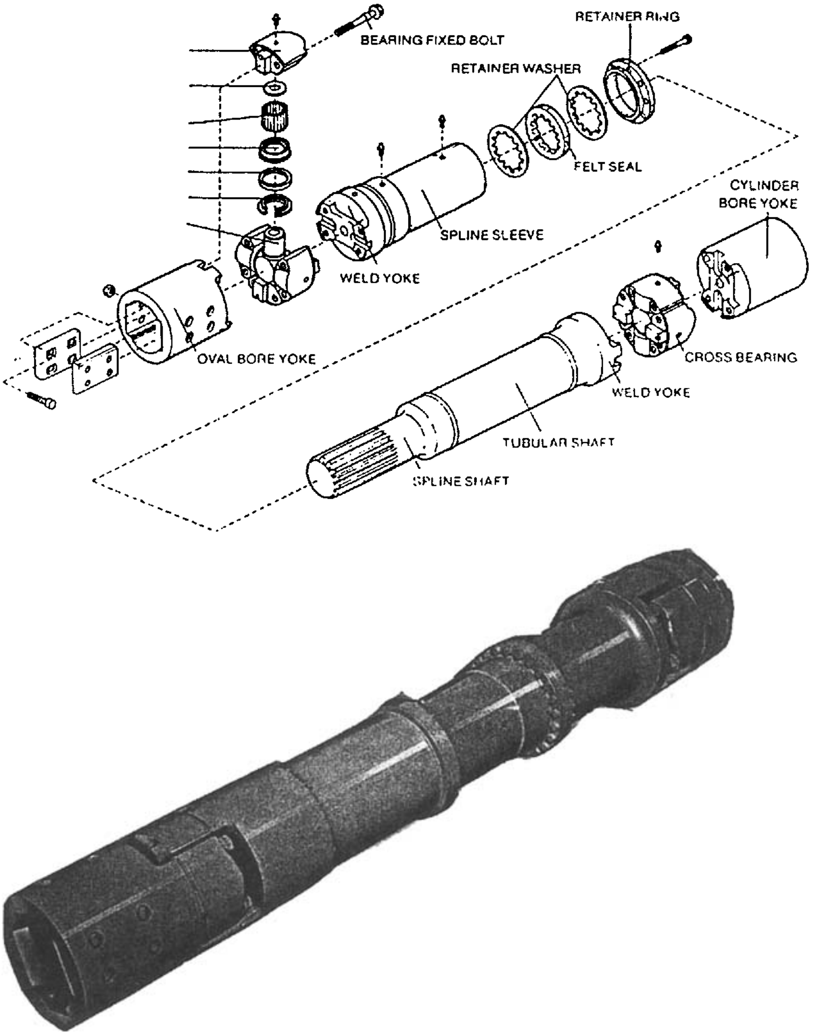


FIGURE 2.45 Block-type U-joint. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

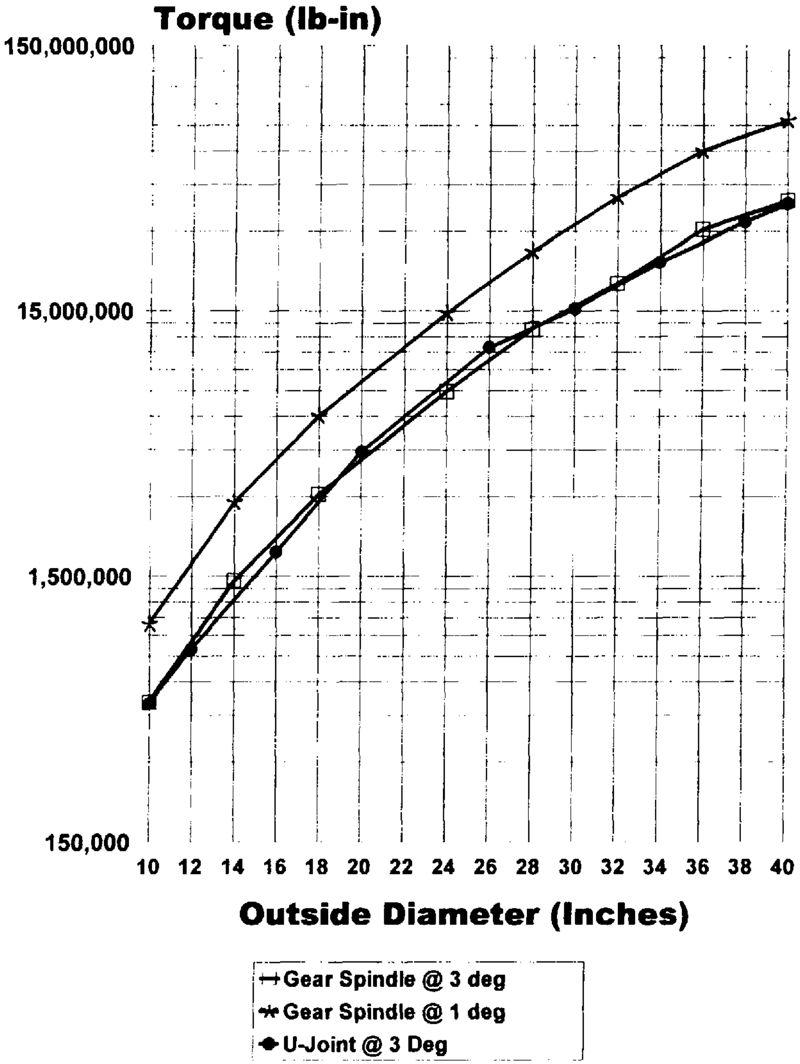


FIGURE 2.46 Torque vs. outside diameter for gear spindles and U-joints.

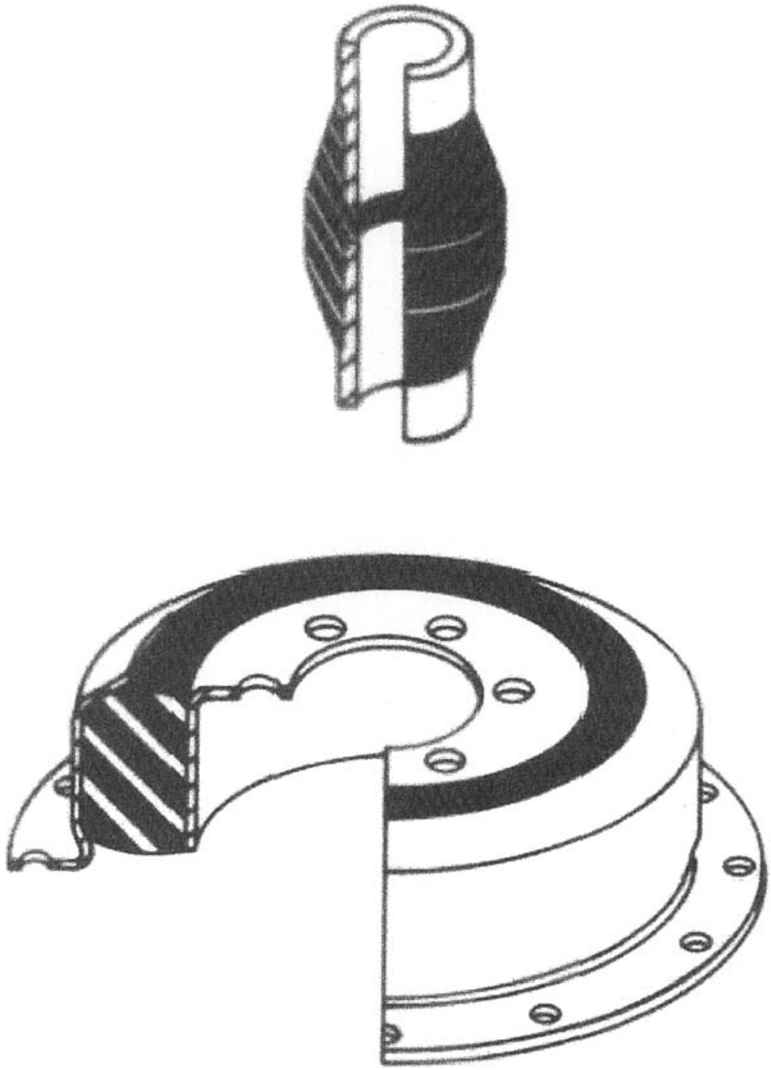


Figure 2.47 Bonded shear coupling. (Courtesy of Lord Corporation, Industrial Products Division.)

diesel drives). Couplings of this type can be very large (over 60 in.). They tend to be 30–50% larger than elastomeric compression couplings that handle the same torque. They are two or three times as flexible as compression-type elastomeric couplings. If the elastomer fails, the coupling totally disengages. These couplings can provide up to 1/2° of misalignment and parallel offset capabilities of 1/32–1/16 in.

2. Elastomeric Block Couplings

Elastomeric block couplings (see Figure 2.48) are used on large critical equipment (such as synchronous motor-driven equipment) and in special-purpose applications to reduce vibratory torque or, because they are torsionally soft, to tune a system. Block couplings use rubber in compression. The rubber blocks are installed in cavities formed by internal

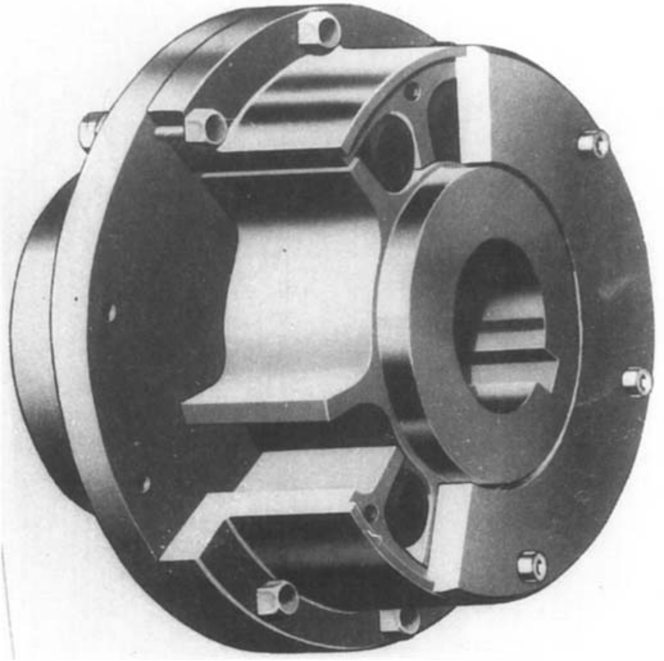


FIGURE 2.48 Block coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

sleeve blades, an external hub blade, and two end plates. This type is unique among couplings due to its fail-safe feature. If the elastomeric element fails, the coupling may run for some time on the metal blades. These couplings can provide up to 1/4° of misalignment and parallel offset capabilities of 1/64–1/32 in.

C. Metallic Element Couplings

1. Disc Couplings

The disc coupling is available in a number of forms (Figure 2.49), all having driving and driven bolts on the same bolt circle. The amount of misalignment that each type can handle depends on the length of material between bolts. Torque is transmitted by driving bolts pulling driven bolts with disc material. More bolts provide greater capacity but reduce the coupling flexibility. For special-purpose applications the discs are provided as a pack. Some disc packs are factory installed, while others are supplied with pilot rings for installation in the field. They are avail-

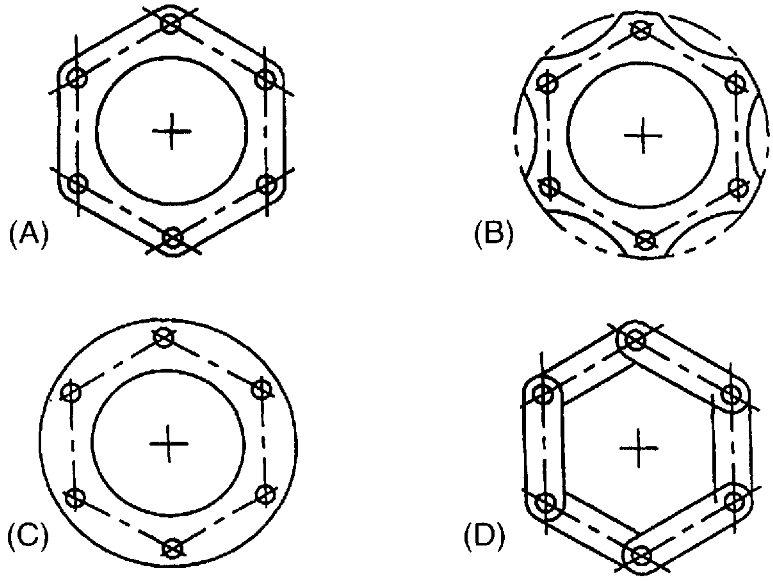


FIGURE 2.49 Various disc shapes.

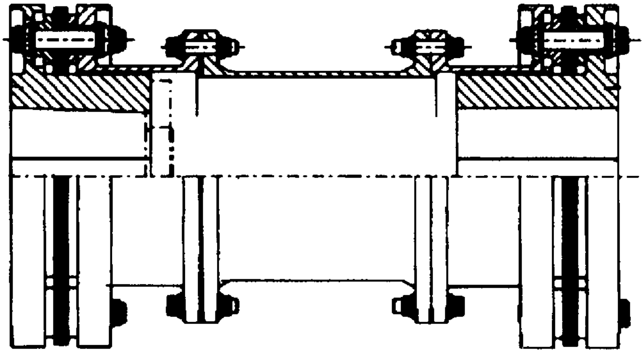


Figure 2.50 Reduced moment disc coupling (scalloped). (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

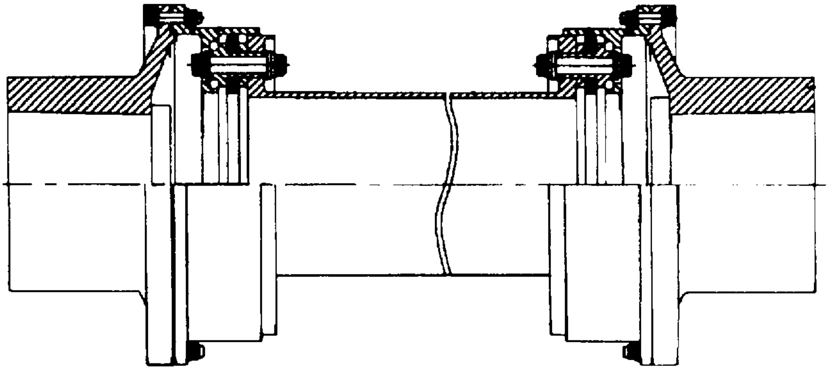


Figure 2.51 Marine style disc coupling (scalloped). (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

able in many styles, including reduced moment (Figure 2.50) and marine (Figure 2.51). High speed and high angles are common for the main drive and tail rotator drive of helicopters. These applications require very flexible couplings like the flexible frame disc coupling (see Figure 2.52).

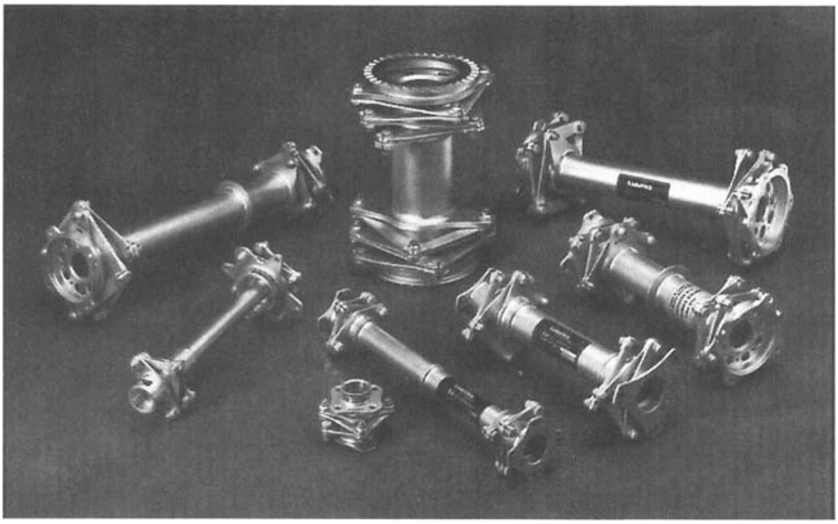


Figure 2.52 Frame disc coupling. (Courtesy of Kamatics Corporation, Kaflex Couplings)

2. Diaphragm Couplings

The diaphragm coupling comes in two basic forms: a single tapered profile, which can be either a welded diaphragm (see Figure 2.53) or a one-piece diaphragm (see Figure 2.54), and a multiple modified profile, which can be either straight with cutouts (see Figure 2.55) or contoured (see Figure 2.56). Both forms have profile modification to reduce size, increase flexibility, and control stress concentration. The torque path passes through the diaphragm member in the radial direction from the outer diameter to the inner diameter. These couplings are used in most special-purpose applications and are available in many shapes and styles, including marine (see Figures 2.53 and 2.55) and reduced moment (see Figure 2.56).

D. Miscellaneous Types of Couplings

1. Offset Couplings (Schmidt Couplings)

Offset couplings (see Figure 2.57) are designed to handle large parallel shafts offset up to 18 in. and torques of up to 1,000,000 lb-in. They are

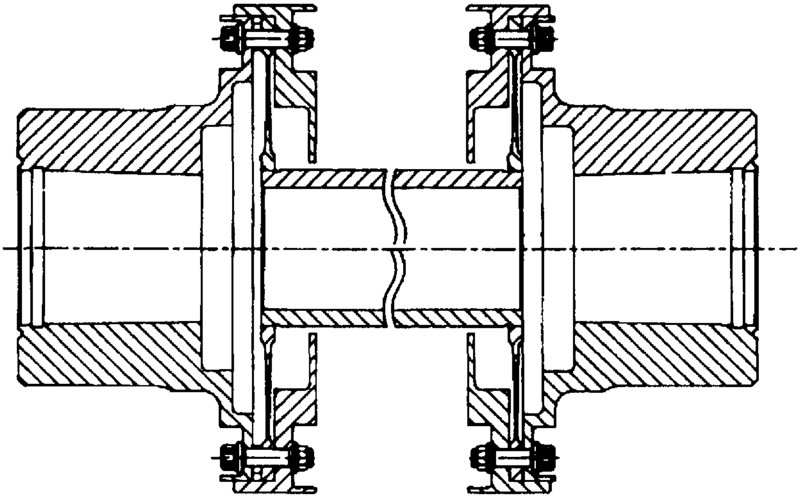


FIGURE 2.53 Tapered diaphragm coupling (welded). (Courtesy of Lucas.)

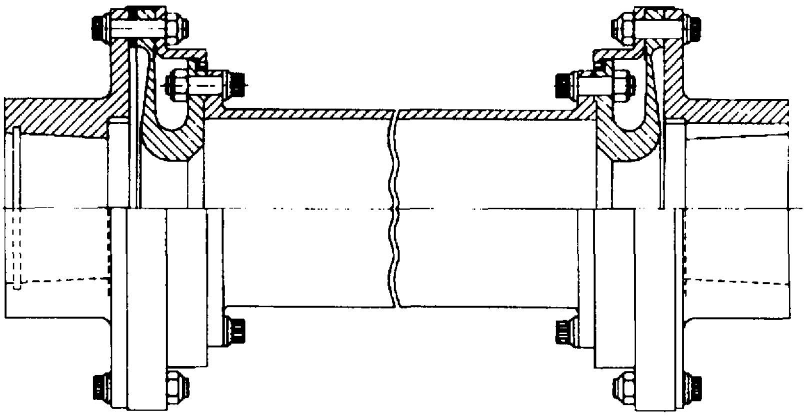


FIGURE 2.54 One-piece diaphragm coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

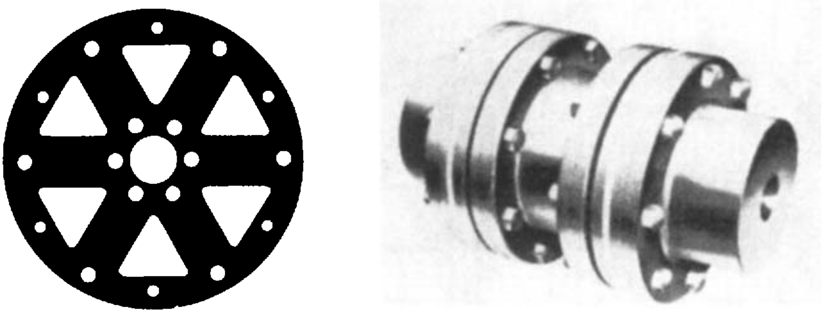


Figure 2.55 Multiple straight diaphragm coupling. (Courtesy of Flexibox International, Metastream Couplings.)

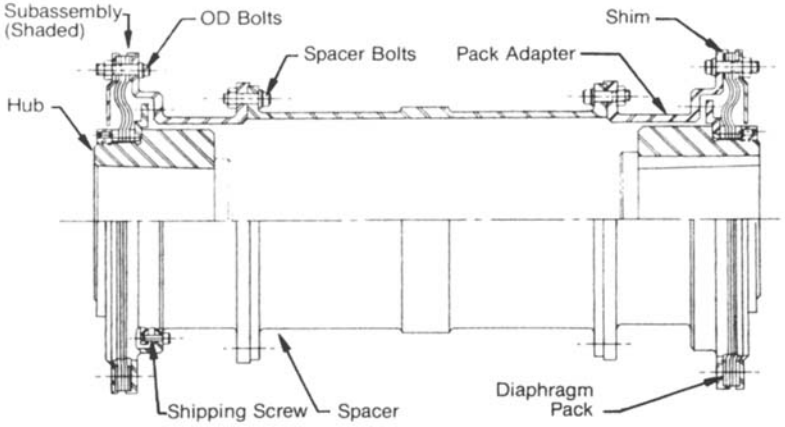


Figure 2.56 Multiple convoluted diaphragm coupling. (Courtesy of Ameridrives International.)

used on steel mill equipment, conveyors, and other heavy duty equipment.

2. Tangential Spring Couplings

Tangential spring couplings (see Figure 2.58) transmit torque through tangentially arranged compression springs. They are used to protect the connected equipment from high shock loads. Tangential spring couplings are used on small to very large horsepower applications (torques to 15,000,000 lb-in).

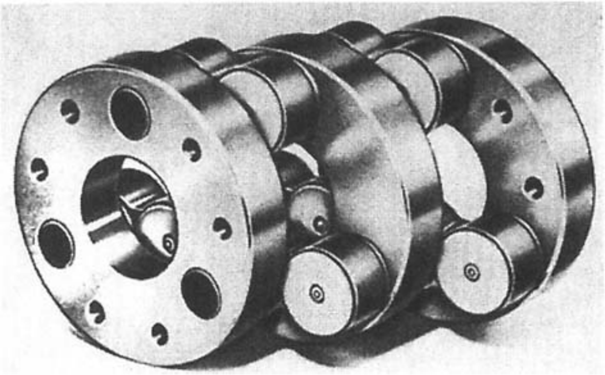


FIGURE 2.57 Offset coupling (Schmidt coupling). (Courtesy of Zero-Max, Motion Control Products.)

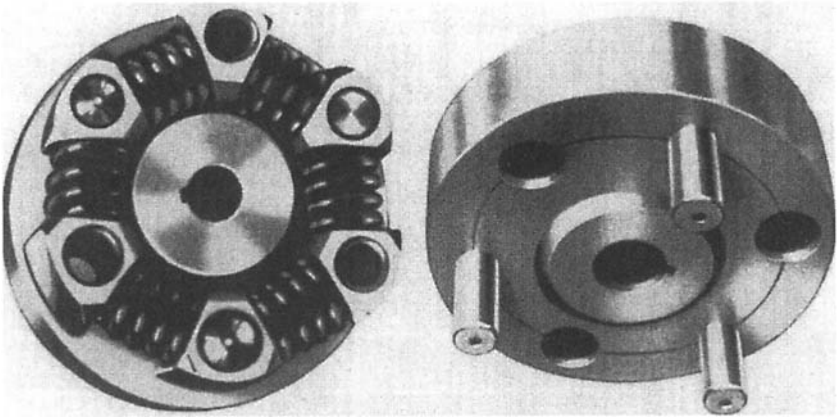


FIGURE 2.58 Tangential spring coupling. (Courtesy of Rexnord, Thomas Couplings.)

V. CHARACTERISTICS AND CAPACITIES OF COUPLINGS

The coupling selector (equipment designer or system designer) must decide what coupling or U-joint is best for the system. The designer must review the possible candidates for a flexible coupling or U-joint and make a selection. The person responsible for the selection of couplings should build a file of the most recent coupling and U-joint catalogs. (See Appendix A for the names and addresses of coupling manufacturers.) This file should be reviewed at regular intervals because designs, models, materials, and ratings are constantly being updated and improved.

Couplings and U-joints are usually selected on the basis of their capacities and characteristics. The two most important capabilities are torque and speed. Figure 2.59 is a graph that defines the torque–speed envelopes of the most common types of couplings. In addition one

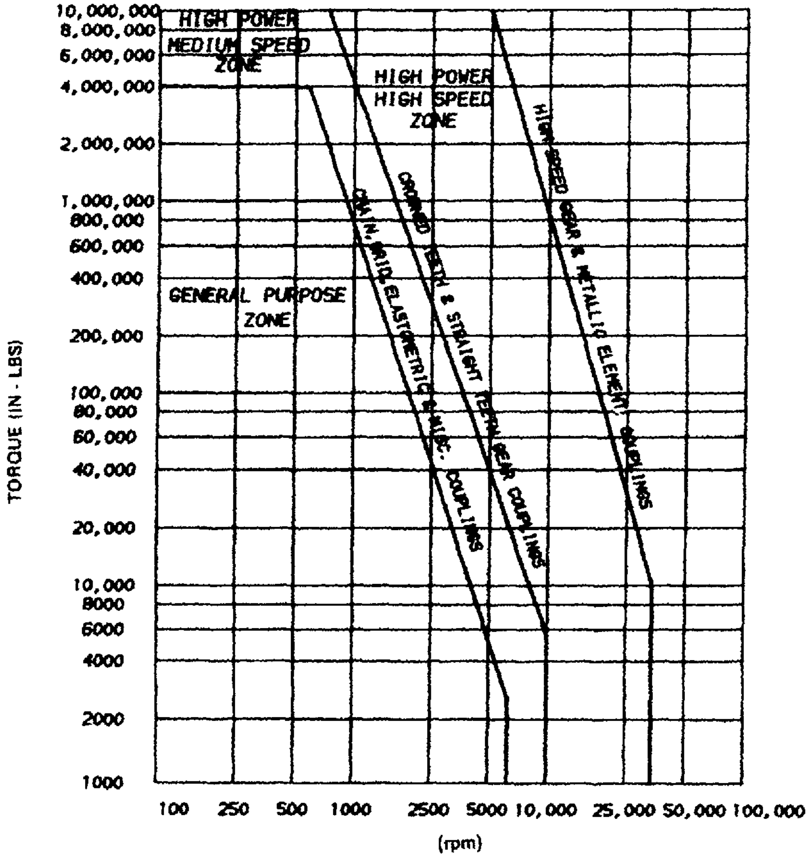


FIGURE 2.59 Torque versus speed (rpm) envelope of various types of couplings.

should consider the capacities of the couplings by reviewing the application requirements and determine which couplings meet the requirements. (See Figure 2.60A for general-purpose couplings and Figure 2.60B for special-purpose couplings.) Finally, the designer must also look at the application and its history and then trade off features and characteristics of available couplings and pick the one best suited for the equipment and operating condition. (See Figure 2.61A for general-purpose couplings and Figure 2.61B for special-purpose couplings.)

GENERAL PURPOSE COUPLINGS	COUPLING FUNCTIONAL CAPACITIES				
	MAX CONTINUOUS TORQUE (IN-LBS)	MAX BORE (IN)	ANGULAR MISALIGNMENT (DEGREE)	PARALLEL OFFSET (IN/IN) or (IN X OD) or (IN)	AXIAL TRAVEL (IN)
MECHANICALLY FLEXIBLE TYPE					
GEAR TYPE					
STRAIGHT TOOTH	54,000,000	45	1/2	.009 IN/IN	1/8-1
CROWNED TOOTH	54,000,000	45	1 1/2	0.026 IN/IN	1/8-1
CHAIN COUPLING					
STEEL	1,350,000	10.5	1-2	0.004 IN/IN	1/4
NYLON	50,000	8	1-2	009 IN/IN	1/2
GRID TYPE					
VERTICAL SPLIT COVER	4,000,000	10	1/3	0.004 IN/IN	3/16
HORIZONTAL SPLIT COVER	4,000,000	20	1/3	0.004 IN/IN	3/16
U-JOINT TYPE					
VEHICLE	30,000	6	15	25 IN/IN	12[a]
MECHANISM	30,000	2.5	35	57 IN/IN	NA
YOKE	135,000	6.5	20	.34 IN/IN	12[a]
ELASTOMERIC ELEMENT TYPE					
SHEAR TYPE					
URETHANE TIRE	175,000	8	4	3/16 IN	1/8
CORDED TIRE	450,000	11	4	1/8 IN	1/4
SHEAR DONUT	75,000	6	1	1/4 IN	5/16
COMPRESSION TYPE					
COMPRESSION DONUT	26,500	5.5	1	0.086	1/16
BLOCK	20,000,000	34	1/4	0.0003 X OD	1/32
JAW	550,000	10	1/4	0.0003 X OD	1/16
PIN & BUSHING	1,000,000	12	1/8	0.0003 X OD	1/16
METALLIC ELEMENT TYPE					
DISC TYPE					
CIRCULAR DISC	4,000,000	15.5	1/2	009 IN/IN	3/8
SQUARE DISC	4,000,000	15.5	1/2	.009 IN/IN	3/8
SCALLOPED DISC	4,000,000	15.5	1/2	.009 IN/IN	3/8
LINK DISC	250,000	7.5	1/2	009 IN/IN	1/4
MISCELLANEOUS TYPE					
PIN & BUSHING	150,000	6	1/8	0 001 x OD	1/8
METALLIC BEAM	20,000	3	1/2	.009 IN/IN	1/4
SLIDING BLOCK	20,000	3 5	1	0.001 x OD	1/8

[a] telescoping shaft is used to accommodate axial travel NA = NOT APPLICABLE

FIGURE 2.60 (A) Coupling functional capacities (general-purpose couplings).

SPECIAL PURPOSE COUPLINGS	COUPLING FUNCTIONAL CAPACITIES				
	MAX. CONTINUOUS TORQUE (IN-LBS)	MAX BORE (IN)	ANGULAR MISALIGNMENT (DEGREE)	PARALLEL OFFSET (IN/IN) OT (1N X OD) OR (IN)	AXIAL TRAVEL (IN)
MECHANICALLY FLEXIBLE TYPE					
GEAR TYPE					
HIGH ANGLE (GEAR SPINDLE)	85,000,000	26	6	0.1 IN/IN	12
HIGH SPEED (SEALED LUBED)	10,000,000	18	1/2	0.009 IN/IN	1
HIGH SPEED (CONTINUOUSLY LUBED)	10,000,000	18	1/2	0.009 IN/IN	1
HIGH SPEED (CONTINUOUSLY LUBED MARINE STYLE)	10,000,000	18	1/2	0.009 IN/IN	1
U-JOINT TYPE					
CONSTANT VELOCITY	56,000	4	20	.34 IN/IN	NA
YOKE	70,000,000	30	15	.25 IN/IN	12*
BLOCK	70,000,000	30	15	.25 IN/IN	12*
ELASTOMERIC ELEMENT TYPE					
SHEAR TYPE					
BONDED SHEAR	350,000	16	1/2	0.0005 X OD	1/16
COMPRESSION TYPE					
BLOCK	20,000,000	34	1/4	0.0003 X OD	1/32
METALLIC ELEMENT TYPE					
DISC TYPE					
REDUCED MOMENT (SCALLOPED)	3,500,000	9	1/3	.006 IN/IN	.375
MARINE STYLE (SCALLOPED)	3,500,000	18	1/3	.006 IN/IN	.5
FRAME	100,000	6	1	0.018	.25
DIAPHRAGM TYPE					
TAPER	6,000,000	20	1/2	0.009 IN/IN	1
ONE PIECE	6,000,000	20	1/2	0.009 IN/IN	1
MULTIPLE STRAIGHT	6,000,000	20	1/2	0.009 IN/IN	1
MULTIPLE CONVOLUTED	6,000,000	20	1/2	0.009 IN/IN	1
MISCELLANEOUS TYPE					
OFFSET (SCHMIDT)	1,000,000	8	NA	18^0	NA
TANGENTIAL SPRING	15,000,000	20	1/8	0.002 IN/IN	NA

* telescoping shaft is used to accommodate axial travel   NA = NOT APPLICABLE

FIGURE 2.60 (B) Coupling functional capacities (special-purpose couplings).

GENERAL PURPOSE COUPLINGS	COUPLING EVALUATION CHARACTERISTICS												
	AXIAL FORCES	TORQUE CAPACITY TO OD	HIGH SPEED CAPACITY	INHERENT BALANCE	BENDING MOMENT	FLEXIBLE ELEMENTS EASILY REPLACED	TORSIONAL STIFFNESS	DAMPING	BACKLASH	LUBRICATION	EASE OF ASSEMBLY	RELATIVE COST	LIFE OF FLEXIBLE ELEMENT
MECHANICALLY FLEXIBLE TYPE													
GEAR TYPE													
STRAIGHT TOOTH	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
CROWNED TOOTH	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
CHAIN COUPLING													
STEEL	M-H	M	L	F	H	Y	M	N	H	Y	E	L-M	1-3
NYLON	L-M	M	L	F	H	Y	M	N	H	Y	E	L-M	1-3
GRID TYPE													
VERTICAL SPLIT COVER	M	M	M	G	M	Y	L-M	L-M	M	Y	E	L-M	2-3
HORIZONTAL SPLIT COVER	M	M	M	G	M	Y	L-M	L-M	M	Y	E	L-M	2-3
U-JOINT TYPE													
VEHICLE	H	L	L	F	H	N	L-M	N	N (a)	Y	G	L	3-5
MECHANISM	H	L	NA	F	H	N	L-M	N	N (a)	Y	G	L	1-3
YOKE	H	M	L	F	H	N	L-M	N	N (a)	Y	G	L-M	1-3
ELASTOMERIC ELEMENT TYPE													
COMPRESSION TYPE													
COMPRESSION DONUT	L-M	L	L	F-G	L	Y	L	M-H	N	N	E	L	3-5
BLOCK	M	M-H	L	F-G	L-M	Y	L-M	M-H	N-L	N	G	M-H	3-5
JAW	M	M	L	F-G	L-M	Y	L-M	M-H	N-L	N	E	L-M	3-5
PIN & BUSHING	M	M	L	F-G	M	Y	M	L-M	N-L	N	G	M	3-5
SHEAR TYPE													
URETHANE TIRE	S	L	L	F	L	Y	L	H	N	N	G	L	2-3
CORDED TIRE	S	L	L	F	L	Y	L	M-H	N	N	G	L-M	3-5
SHEAR DONUT	S	L	L	F	L	Y	L	H	N-L	N	G	L	2-3
METALLIC ELEMENT TYPE													
DISC TYPE													
CIRCULAR DISC	L-M	M-H	H	E	L-M	Y	M	N-L	N	N	G	M-H	4-8
SQUARE DISC	L-M	M-H	H	E	L-M	Y	M	N-L	N	N	G	M-H	4-8
SCALLOPED DISC	L-M	M-H	H	E	L-M	Y	M	N-L	N	N	G	M-H	4-8
LINK DISC	L	M-H	H	G	L-M	Y	M	N-L	N	N	F	M-H	4-8
MISCELLANEOUS TYPE													
PIN & BUSHING	H	M	L	F	H	Y	M	N-L	L	N	F	M	1-3
METALLIC BEAM	L-M	L	M	G	L	Y	M	N	N	N	G	M-H	4-8
SLIDING BLOCK	H	L	L	F	H	Y	M	N	M	Y	G	M	1-3

a) No backlash unless telescoping shaft is used to accommodate axial travel

M = Medium
S = Small
H = High
L = Low
NA = Not Applicable
E = Excellent
G = Good
F = Fair
Y = Yes
N = No/None
? = Maybe

FIGURE 2.61 (A) Coupling characteristics (general-purpose couplings).

SPECIAL PURPOSE COUPLINGS	COUPLING EVALUATION CHARACTERISTICS												
	AXIAL FORCES	TORQUE CAPACITY TO OD	HIGH SPEED CAPACITY	INHERENT BALANCE	BENDING MOMENT	FLEXIBLE ELEMENTS EASILY REPLACED	TORSIONAL STIFFNESS	DAMPING	BACKLASH	LUBRIC.	EASE OF ASSEMBLY	RELATIVE COST	LIFE OF FLEXIBLE ELEMENT
MECHANICALLY FLEXIBLE TYPE													
GEAR TYPE													
HIGH ANGLE (GEAR SPINDLE)	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
HIGH SPEED (SEALED LUBED)	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
• • • (CONTINUOUSLY LUBED)	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
• • • (CONTINUOUSLY LUBED MARINE STYLE)	M-H	H	H	G	H	N	H	N-L	M-H	Y	E	M	3-5
U-JOINT TYPE													
CONSTANT VELOCITY	H	L	L	F	H	N	L-M	N-L	Na	Y	E	L	3-5
YOKE	H	H	NA	F	H	N	L-M	N-L	Na	Y	E	L	1-3
BLOCK	H	H	L	F	H	N	L-M	N-L	Na	Y	E	L-M	1-3
ELASTOMERIC ELEMENT TYPE													
SHEAR TYPE													
BONDED SHEAR	L	L	L	F	L	Y	L	H	N	N	E	L	3-5
COMPRESSION TYPE													
BLOCK	M	M-H	L	F-G	L-M	Y	M	M-H	N-L	N	G	M-H	3-5
METALLIC ELEMENT TYPE													
DISC TYPE													
REDUCED MOMENT (SCALLOPED)	L-M	M-H	H	E	L-M	Y	M	N-L	N	N	G	M-H	4-8
MARINE STYLE (SCALLOPED)	L-M	M-H	H	E	L-M	Y	M	N-L	N	N	G	M-H	4-8
FRAME	L-M	M-H	H	E	L-M	?	M	N-L	N	N	G	M-H	4-8
DIAPHRAGM TYPE													
TAPERED	L-M	M-H	H	E	L-M	?	M-H	N-L	N	N	G	M-H	4-8
ONE PIECE	L-M	M-H	H	E	L-M	?	M-H	N-L	N	N	G	M-H	4-8
MULTIPLE STRAIGHT	L-M	M-H	H	E	L-M	?	M-H	N-L	N	N	G	M-H	4-8
MULTIPLE CONVOLUTED	L-M	M-H	H	E	L-M	?	M-H	N-L	N	N	G	M-H	4-8
MISCELLANEOUS TYPE													
OFFSET (SCHMIDT)	H	M	L	F	H	Y	M	L	L	N	F	M	1-3
TANGENTIAL SPRING	L-M	L	M	G	L	Y	M	L-M	N	N	G	M-H	4-8

a) No backlash unless telescoping shaft is used to accommodate axial travel

M = MEDIUM
S = SMALL
H = HIGH
L = LOW
NA= NOT APPLICABLE
E = EXCELLENT
G = GOOD
F = FAIR
Y = YES
N = NO/NONE
? = MAYBE

FIGURE 2.61 (B) Coupling characteristics (special-purpose couplings).

This page intentionally left blank

3

Selection and Design

I. SELECTION FACTORS

Flexible couplings are a vital part of a mechanical power transmission system. Unfortunately, many system designers treat flexible couplings as if they were just pieces of hardware. The amount of time spent in selecting a coupling and determining how it interacts with a system should be a function not only of the cost of the equipment but also of how much downtime it will take to replace the coupling or repair a failure. In some cases the process may take only a little time and be based on past experience; however, on sophisticated systems it may require complex calculations, computer modeling, finite element analysis (FEA), and possibly even testing.

A system designer or coupling user cannot put just any flexible coupling into a system with the hope that it will work. It is the system designer's or user's responsibility to select a coupling that will be compatible with the system. Flexible coupling manufacturers are usually not system designers; rather, they size, design, and manufacture couplings to fulfill the requirements supplied to them. Some coupling manufacturers can offer assistance based on past experience. They are the experts in the design of flexible couplings, and they use their know-how in design, materials, and manufacturing to supply a standard coupling or a custom-built one if requirements so dictate. A flexible coupling is usually the least expensive major component of a rotating system, generally accounting for less than 1% and probably never exceeding 10% of the total system cost.

Flexible couplings must accommodate the calculated loads and forces imposed on them by the equipment and their support structures. These may be due to thermal expansion, torque-load variations, or many other conditions. In operation, flexible couplings are sometimes called on to accommodate for the unexpected. These may be forgotten conditions or forces unleashed when the coupling interacts with the system, which may precipitate coupling or system problems. It is the coupling selector's responsibility to review the interaction of the coupling with the system it's to be used in. When this interaction is forgotten in the selection process, coupling life may be shortened during operation or a costly failure of coupling or equipment may occur.

A. Selection Steps

There are usually four steps that a coupling selector should take to ensure the proper selection of a flexible coupling for critical or vital equipment.

1. Review the initial requirements for a flexible coupling and select the type of coupling that best suits the system.
2. Supply the coupling manufacturer with all the pertinent information on the application so that the coupling can be properly sized, designed, and manufactured to fit the requirements.
3. Obtain the flexible coupling's characteristic information so the interactions of the coupling with the system can be checked to ensure compatibility and prevent the unleashing of detrimental forces and moments.
4. Review the interactions, and if the system conditions do change, contact the coupling manufacturer so that the new

conditions and their effect on the coupling selected can be reviewed. Continue this process until the system and the coupling are compatible.

The complexity and depth of the selection process for a flexible coupling depend on how critical and how costly downtime would be to the ultimate user. The coupling selector has to determine to what depth the analysis must go. As an example, a detailed, complex selection process would be abnormal for a 15 hp motor-pump application but would be normal for a multimillion dollar 20,000 hp motor, gear, and compressor train.

B. Types of Information Required

Coupling manufacturers will know only what the coupling selector tells them about an application. Missing information required to select and size couplings can lead to an improperly sized coupling and possible failure. A minimum of three items are needed to size a coupling: horsepower, speed, and interface information. This is sometimes the extent of the information supplied or available. For coupling manufacturers to do their best job, the coupling selector should supply as much information as is felt to be important about the equipment and the system. Figure 3.1 lists the types of information required. Figures 3.2 and 3.3 list specific types of information that should be supplied for the most common types of interface connections: the flange connection and the cylindrical bore.

C. Interaction of a Flexible Coupling with a System

A very important but simple fact often overlooked by the coupling selector is that couplings are connected to the system. Even if they are selected, sized, and designed properly, this does not ensure trouble-free operation. Flexible couplings generate their own forces and can also amplify system forces. This may change the system's original characteristics or operating conditions. The forces and moments generated by a coupling can produce loads on equipment that can change the alignment, decrease the life of a bearing, or unleash peak loads that can damage or cause failure of the coupling or connected equipment.

Listed below are some of the coupling characteristics that may interact with the system. The coupling selector should obtain the values

1. Horsepower
2. Operating speed
3. Interface connect information in Figs. 3.1, 3.2, etc.
4. Torques
5. Angular misalignment
6. Offset misalignment
7. Axial travel
8. Ambient temperature
9. Potential excitation or critical frequencies
   a. Torsional
   b. Axial
   c. Lateral
10. Space limitations (drawing of system showing coupling envelope)
11. Limitation on coupling-generated forces
    a. Axial
    b. Moments
    c. Unbalance
12. Any other unusual condition or requirements or coupling characteristics—weight, torsional stiffness, etc.

Note: Information supplied should include all operating or characteristic values of equipment for minimum, normal, steady-state, momentary, maximum transient and the frequency of their occurrence.

FIGURE 3.1 Types of information required to properly select, design, and manufacture a flexible coupling.

for these characteristics from the coupling manufacturer and analyze their effect on the system.

- Torsional stiffness
- Torsional damping
- Amount of backlash
- Weights
- Coupling flywheel effect (defined on p. 79)
- Center of gravity
- Amount of unbalance
- Axial force

1. Size of bore including tolerance or size of shaft and amount of clearance or interference required
2. Lengths
3. Taper shafts
   a. Amount of taper
   b. Position and size of O-ring grooves if required.
   c. Size, type, and location of hydraulic fitting
   d. Size and location of oil distribution grooves
   e. Maximum pressure available for mounting
   f. Amount of hub draw-up required
   g. Hub O.D. requirements
   h. Torque capacity required (should also specify the coefficient of friction to be used)
4. Minimum strength of hub material or its hardness
5. If keyways in shaft
   a. How many
   b. Size and tolerance
   c. Radius required in keyway (minimum and maximum).
   d. Location tolerance of keyway respective to bore and other key ways

FIGURE 3.2 Information required for cylindrical bores.

1. Diameter of bolt circle and true location
2. Number and size of bolt holes
3. Size, grade, and types of bolts required
4. Thickness of web and flange
5. Pilot dimensions

FIGURE 3.3 Types of interface information required for bolted joints.

- Bending moment
- Lateral stiffness
- Coupling axial critical frequency (discussed below)
- Coupling lateral critical frequency (discussed below)

The *flywheel effect* of a coupling is the product of the coupling weight times the square of the radius of gyration (WR^2). The radius of gyration is that radius at which the mass of the coupling can be considered to be concentrated.

Coupling manufacturers usually calculate coupling *axial* and *lateral critical values* with the assumption that the equipment is infinitely rigid. A coupling's axial and lateral critical values express a coupling's vibrational natural frequency in terms of rotational speed (revolutions per minute, rpm).

D. Final Check

The coupling selector should use the coupling characteristics to analyze the system axially, laterally, thermally, and torsionally. Once the analysis has been completed, if the system's operating conditions change, the coupling selector should supply this information to the coupling manufacturer to ensure that the coupling selection has not been sacrificed and that a new coupling size or type is not required. This process of an exchange of information between the coupling manufacturer and the coupling selector should continue until the system and coupling are compatible. It is the only way to ensure that the system will operate successfully.

II. DESIGN EQUATIONS AND PARAMETERS

The intent of this section is to give the reader (coupling selector or user) some basic insights into coupling ratings and design. The equations and allowable values set forth in this section should be used only for comparison, not for design. They will help the reader compare "apples to apples"—in this case, couplings to couplings.

A. Coupling Ratings

1. Torque Ratings

It has become more and more confusing as to what a factor of safety (F.S.) and a service factor (S.F.) (also application or experience factor) really are. Many people use the terms. "service factor" and "factor of safety" interchangeably. There is an important distinction, however, and an understanding of the difference is essential to proper coupling selection for a particular application.

a. Factor of Safety. Factors of safety are used in the *design* of a coupling. Coupling designers use factors of safety because there are uncertainties in the design. The designer's method of analysis uses approximations to model the loading, and therefore the calculated stresses may not be exact. Likewise, material properties such as modulus, ultimate strength, and fatigue strength have associated tolerances that must be considered.

Today, with the use of such computational tools as FEA, stress analysis is generally capable of more accurate results than in the past. In addition, the properties of the materials used in most coupling components are more controlled and better known. Therefore, compared with those designed 20 years ago, couplings designed today can indeed operate safely with lower calculated factors of safety. Also, the design safety factor of some couplings such as flexible element couplings can be lower than for other types of couplings simply because the "safeness" is more accurately predicted.

Some modern couplings have stress loading that is more easily determined. The stresses from misalignment, axial displacement, and torque are generally more accurately known than is the case with other couplings. Some couplings (such as gear couplings) have a number of variables that affect their design (such as tooth form, surface finish, materials, temperature, and lubrication). Life and safeness are difficult to evaluate for many mechanically flexible and elastomeric couplings.

Generally, torque is the most significant contributor of load to the overall stress picture. The fatigue factor of safety of a flexible element coupling is generally not as affected by torque, because the failure mode in these couplings is not very sensitive to torque during continuous operation.

b. Service Factor. A service factor is used to account for the higher operating torque conditions of the equipment to which the coupling is connected. A service (or experience) factor should be applied to the normal operating torque of, for instance, a turbine or compressor. This factor accounts for torque loads that are not normal but may be encountered continuously, such as low-temperature driver output, compressor fouling, or possible vibratory torques. Service factors are also sometimes used to account for the real operating conditions, which may be 5–20% above the equipment rating.

Different service factors are used or recommended depending on the severity of the application. Is it a smooth-running gas turbine-driven compressor application, or will the coupling be installed on a reciprocating pump? Also remember that service factors should be applied to continuous operating conditions rather being used to account for starting torques, short-circuit conditions, rotor rubs, and similar transient conditions.

2. Design and Selection Criteria Terms

A factor of safety (F.S.) is used to cover uncertainties in a couplings *design*—analytical assumptions in stress analysis, material unknowns, manufacturing tolerances, etc. Under given design conditions the FS is the ratio of strength (or stress capacity) to predicted stress, where the stress is a function of torque, speed, misalignment, and axial displacement.

A *design factor of safety* (D.F.S.) is the factor of safety at the catalog rated conditions of torque, speed, misalignment, and axial displacement. It is used by the manufacturer to establish the coupling rating, because it is the maximum loading that the manufacturer says the coupling can safely withstand.

The factor of safety of most interest to us, however, is the factor of safety at the particular set of application loads to which the coupling is continuously subjected. We define this to be an *application factor of safety* (A.F.S.). In fact, the application factor of safety is the measure of safety that would allow the coupling selector to evaluate the coupling's safeness under actual operation and compare various couplings.

A service factor (application factor or experience factor) (S.F.) is normally specified by the purchaser (although assistance is sometimes given by the manufacturer). It is a *torque multiplier.* It is applied to the operating torque (called the "normal operating point" in API 671) of the connected equipment. The service factor torque multiplier is used to account for torque loads that are beyond the normal conditions and are of a recurring nature. Couplings are generally selected by comparing the selection torque (S.F. × normal operating torque) to the coupling's maximum continuous rating. Service factors account for conditions such as compressor fouling, changes in the molecular weight, temperature, or pressure of the pumped fluid, or changes in any other repetitive loading conditions that may occur over 10^6 revolutions of the coupling. And sometimes service factors are used to account for the real operating conditions of the equipment, which may be 5–20% above the equipment rating. Service factors should not be applied to account for starting torques or short-circuit torques, although these conditions are sometimes stated as a multiple of normal torque.

The *endurance limit* is the failure strength limit of a coupling component subjected to a combination of constant and alternating stresses. Beyond this limit the material can be expected to fail after some finite number of cyclic loads. Below this limit the material can be expected to have infinite life (or a factor of safety greater than 1.0).

The yield limit (Y.L.) is determined by the manufacturer to be the failure strength limit of a coupling component that *will cause detrimental damage.* If this limit is exceeded, the coupling should be replaced.

The *coupling rating* is a torque capacity at rated misalignment, axial displacement, and speed. This applies to the ratings given below.

The *maximum continuous rating* (M.C.R.) is determined by the manufacturer to be the torque capacity at which a coupling can safely run continuously and have an acceptable design factor of safety.

The *peak rating* (P.R.) is determined by the manufacturer to be the torque capacity that a coupling can handle without experiencing localized yielding of any of its components. Additionally, a coupling can handle this torque condition for 5000–10,000 cycles without failing.

The *maximum momentary rating* (M.M.R.) is determined by the manufacturer to be the torque capacity that a coupling can experience without ultimate failure although localized yielding (damage) of one of its components may occur. A coupling can withstand this occurrence once for a brief time. After that, the coupling should be inspected and possibly replaced. (This is also sometimes called the *short-circuit torque rating.*)

3. Continuous Operating Conditions and Fatigue Factors of Safety

We will use the flexible element as an example of how an endurance factor of safety is calculated. This type of analysis can be used on any metallic torque-transmitting components of any coupling (bolts, spacers, hubs, etc.)

ANALYSIS OF A FLEXIBLE ELEMENT

The diaphragm, diaphragm pack, or disc pack is the heart of a flexible element coupling and in general is the most highly stressed component during continuous operation. It must accommodate the constant (steady-state or mean) stresses from axial displacement, torque, and centrifugal forces while also withstanding the alternating (cyclic) stresses from angular misalignment and possibly alternating torques. Note that normally other components of the coupling such as flanges, tubes, and bolts are not subjected to the same magnitudes and types of stresses.

To analyze a flexible element and determine its (and generally the coupling's) application factor of safety (AFS) at different loading conditions, its endurance limit must be determined. The problem here, though, is, What failure criteria should be used to determine this limit? What assumptions are made in combining the stresses? Once a criterion is selected, how is the factor of safety determined? What is an appropriate factor of safety for a particular type of coupling?

Let us consider the following load conditions and stresses for, let's say, a diaphragm coupling in a turbine-driven compressor application. (Note that the stresses presented here are for illustrative purposes only.)

Condition	Amount	Stress
Torque	400,000 in.-lb	42,000 psi: constant, shear stress
Speed	13,000 rpm	12,000 psi: constant, biaxial stress
Axial displacement	±0.120 in.	35,000 psi: constant, biaxial stress
Angular misalignment	±0.25°	17,000 psi: alternating, biaxial stress

To calculate the fatigue factor of safety, there are four basic steps:

1. Determine the basic *normal stresses* that result from the stated operating conditions.
2. Apply an appropriate *failure theory* to represent the combined state of stress.
3. Apply an appropriate *fatigue failure criterion* to establish an equivalent mean and an equivalent cyclic stress with which to compare the material fatigue strength.
4. Calculate the factor of safety.

First, the way in which the stresses in this example were determined is subject to evaluation. Various methods may be employed to determine the normal stresses. These methods include classical solutions, empirical formulas, numerical methods, and finite element analysis (FEA). The accuracy of each of these methods is largely dependent on the loading assumptions made in the analysis.

Second, after the fluctuating normal stresses are calculated, they must be combined to provide an accurate representation of the biaxial state of stress by applying an appropriate *failure theory*. The most accurate choice of theory is generally a function of material characteristics and the type of loading. Among the failure theories that might be employed are maximum principal stress, maximum shear stress, and maximum distortion energy (von Mises).

Third, after an appropriate failure theory has been applied, an equivalent constant and an equivalent alternating stress must be determined by applying an appropriate *fatigue failure criterion*. The possible choices here include the Soderberg criterion, the Goodman criterion, the modified Goodman criterion, and constant life fatigue diagrams.

Finally, a fatigue factor of safety can be determined by comparing the equivalent stress to the fatigue failure strength. To do this, an assumption must be made as to how the stress increase is most likely to occur. The most common approach is to combine constant and cyclic stresses (proportional increases of all stresses and loads).

For this example (and most coupling components), it would be conservative (and easy to compare apples to apples) to combine the stresses by using the maximum distortion energy failure theory and applying the modified Goodman fatigue failure criteria to obtain combined mean (constant) stress (s_m) of 87,500 psi and a cyclic (alternating) stress (s_a) of 17,000 psi. The endurance strength (S_e) is 88,000 psi, the yield strength (S_{ty}) is 165,000 psi, and the ultimate diaphragm material strength (S_{tu}) is 175,000 psi.

Using a modified Goodman diagram (Figure 3.4) to plot constant and alternating stresses, and then *using the proportional increase method*, a safety factor of 1.44 is found.

$$\frac{1}{N}=\frac{S_a}{S_e}+\frac{S_m}{S_{tu}}$$

$$\frac{1}{N}=\frac{17{,}000}{88{,}000}+\frac{87{,}500}{175{,}000}$$

$$=0.193+0.50$$

$$\frac{1}{N}=0.693$$

$$N=1.44$$

4. Peak and Maximum Momentary Conditions and Factors of Safety

Just as for continuous torque ratings, there are different ways to rate a coupling's capability to handle noncontinuous peak torques or low-fre-

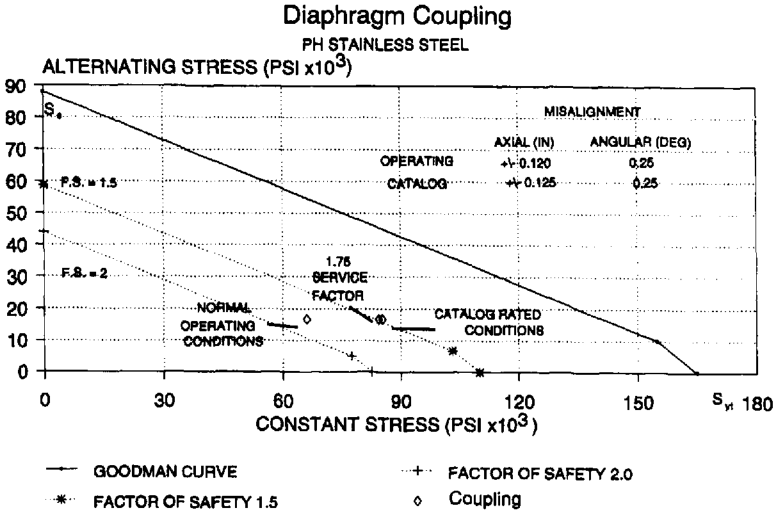


FIGURE 3.4 Modified Goodman diagram.

quency high cyclic torques. These can be caused by such things as motor start-ups, short-circuit conditions, compressor surges, or other transient conditions.

Some questions to ask are: What is the nature of the load? Is it due to a synchronous motor start-up with hundreds of high torque reversals during a daily start-up? Is it a single unidirectional torque induction motor driven application?

As for the coupling, how much capacity above the maximum continuous rating is there before serious damage to the coupling occurs? Some couplings have a catalog peak rating in the range of 1.33–2 times the maximum continuous catalog torque rating. Most of these couplings can handle torques 1.75–2.5 times their normal rating before detrimental damage occurs.

5. Recommended Factors of Safety

For the factor of safety for *maximum continuous rating,* it is recommended that the factor of safety be determined by using the *modified Goodman criteria* (the most widely accepted fatigue failure criteria for steel components). It is further recommended that these factors of safety be calculated using *proportional increase in stress* assumptions (see Figure 3.4).

For *peak rating* and *maximum momentary rating,* the factor of safety is determined by the ratio of yield strength to the stresses calculated at these ratings.

Note that the design factors of safety listed in Table 3.1 are factors for rated conditions only and that the safety factors for a particular application (with or without service factors applied) can generally be expected to be higher.

The lower numbers listed in Table 3.1 have been adopted by API

TABLE 3.1 Design Factors of Safety for Rated Conditions

Coupling capacity	Factors	Basis of safety
Max. continuous rating	1.25–1.35 min	Endurance
Peak rating	1.15–1.25 min	Yield
Max. momentary rating[a]	1–1.10 min	Yield

[a] When the design factor of safety is 1, this may be called the maximum momentary *limit.*

671. These minimum factors and the methods recommended above allow one to properly compare couplings for a particular application. Higher factors might be recommended by the coupling manufacturer if the analysis is not sophisticated and/or the material design limits are not substantiated precisely.

Finally, the values in the table are recommended as a guide and do not reflect how good a job was done in determining and combining the stresses used to obtain them. A certain level of trust is required with each coupling manufacturer based on experience with the product and the organization.

6. Service Factors

Service factors have evolved from experience and are based on past failures. That is, after a coupling failed, it was determined that if the normal operating torque were multiplied by a certain factor and then the coupling was sized accordingly, the coupling would not fail. Since coupling manufacturers use different design criteria, many different service factor charts are in use for the same types of couplings. Therefore, it becomes important when sizing a coupling to follow and use the ratings and service factors recommended in the particular coupling manufacturer's catalog and not intermix them with other manufacturers' procedures and factors. If a coupling manufacturer is told the load conditions—normal and peak loads and their duration—a more detailed and probably more accurate sizing can be developed. Service factors become less significant when the load and duty cycle are known. It is only when a system is not analyzed in depth that service factors must be used. The more that is known about the operating conditions, the closer to unity the service factor can be.

For special-purpose couplings, API 671 defaults to a 1.5 service factor for metallic element couplings and 1.75 for gear and elastomeric couplings that is to be applied to the normal operating torque. *Note:* API cautions that if reasonable attempts to achieve the specified experience factor fail to result in a coupling weight and subsequent overhung moment commensurate with the requirement for the rotor dynamics of the connected machines, a lower factor may be selected by mutual agreement of the purchaser and the vendor. The selected value shall not be less than 1.2.

Most coupling manufacturers recommend service factors for various coupling applications. Some recommend higher values than others. Use the values recommended by the specific coupling manufacturer.

7. Which Coupling Rating Is Correct?

All coupling ratings are correct—as long as you use the recommended service factors and the selection procedure outlined by the coupling manufacturer.

First, two safety factors should be compared:

1. The safety factor related to the endurance strength (for cyclic or reversing loading) or yield strength (for steady-state loading) for normal operating torque and conditions
2. The safety factor related to yield strength for infrequent peak operating torques and conditions

Such a comparison will probably permit the coupling selector to select the safest coupling for the system.

To complete the selection analysis, the coupling selector should obtain the following information from the coupling manufacturer:

1. The material properties of the major components for tensile and shear
    a. Ultimate strength
    b. Yield strength
    c. Endurance limit
2. The stresses in the major components
    a. At normal torque and loads
    b. At peak torque and loads
3. The stress concentration factors (this is important to know when cyclic and reversing torques and/or loading are present)
4. What the manufacturer typically uses or recommends as a safety factor

Most coupling manufacturers will supply this information to the coupling selector. However, coupling manufacturers will generally not supply design equations to calculate the stresses of the flexing element (gear teeth, disc, diaphragms, etc.), since these are usually proprietary.

8. Speed Ratings

The true maximum speed limit at which a coupling can operate cannot be divorced from the connected equipment. This makes it very difficult to rate a coupling because the same coupling can be used on different types of equipment. Let's forget the system for a moment and consider only the coupling.

a. Maximum Speed Based on Centrifugal Stresses. The simplest method of establishing a coupling maximum speed rating is to base it on centrifugal stress (S_t):

$$S_t = \frac{\rho V^2}{g} \tag{3.1}$$

where

ρ = density of material (lb/in.^{3})
V = velocity (in./sec)
g = acceleration (386 in./sec)

$$V = \frac{D_o \times \pi \times \text{rpm}}{60} \tag{3.2}$$

$$\text{rpm} = \frac{375\sqrt{S_t/\rho}}{D_o} \tag{3.3}$$

where D_o is the outside diameter (in.) and

$$S_t = \frac{S_{\text{yld}}}{\text{F.S.}} \tag{3.4}$$

where

F.S. = factor of safety (can be from 1 to 2, typically 1.5)
S_{yld} = yield strength of material (psi)

Material	ρ (lb/in.^{3})
Steel	0.283
Aluminum	0.100

b. Maximum Speed Based on Lateral Critical Speed. For long couplings the maximum coupling speed is usually based on the lateral critical speed of the coupling. Couplings are part of a drivetrain system. If the system is very rigid, the maximum speed might be greater than calculated, and if the system is soft and has a long shaft overhang, the maximum speed may be much lower than the calculated value. Figure 3.5A is a schematic of a coupling connected to a system. Figure 3.5B is a free-body diagram of a mass–spring system that models the coupling connected to the system. The center section of a coupling can be represented as a simply supported beam.

$$N_c = 211.4\sqrt{\frac{1}{\Delta}} \tag{3.5}$$

$$\Delta = \frac{5WL^3}{384EI} \tag{3.6}$$

$$I = 0.049\,(D_o^4 - D_i^4) \tag{3.7}$$

$$K_c = \frac{W}{g}\left(\frac{\pi N_c}{30}\right)^2 \tag{3.8}$$

$$\frac{1}{K_e} = \frac{1}{2(K_c/2)} + \frac{1}{2K_{LL}} \tag{3.9}$$

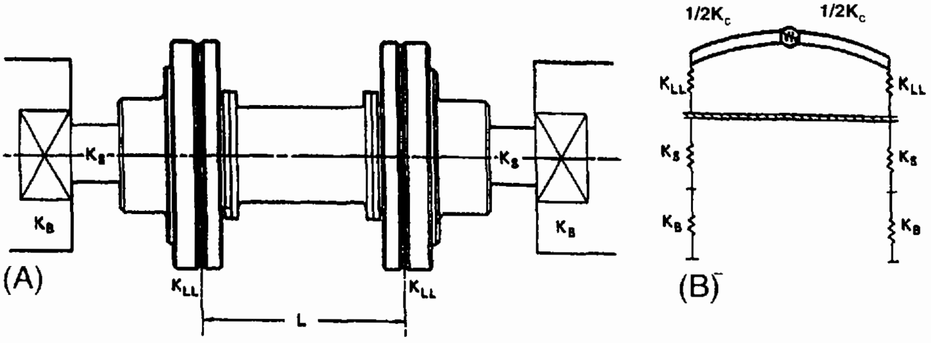


FIGURE 3.5 (A) Schematic of a coupling as connected to a system. (B) Free-body diagram of a coupling as connected to a system.

where

N_c = critical shaft whirl rotational frequency of the coupling shaft [cycles per minute (cpm)]
Δ = shaft end deflection (in.)
L = effective coupling length (in.)
I = moment of inertia of the coupling spacer (in.^{4})
W = center weight (supported) of the coupling (lb)
$E = 30 \times 10^6$ for steel (psi)
D_o = outside diameter of shaft (in.)
D_i = inside diameter of shaft (in.)
g = 386 in./sec^2
K_c = stiffness of coupling spacer (lb/in.)
K_{LL} = lateral stiffness of coupling (lb/in.)
K_e = effective spring rate (lb/in.) for the system shown in Figure 3.5B and calculated using equation (3.10)

These equations ignore the stiffness of the system, which is usually how coupling manufacturers determine critical speed calculations. [*Note:* For some couplings (e.g., gear couplings), K_{LL} can be ignored.]

If we consider the stiffness of the connected shafts and bearings, then

$$\frac{1}{K_e} = \frac{1}{2(K_c/2)} + \frac{1}{2K_{LL}} + \frac{1}{2K_s} + \frac{1}{2K_B} \tag{3.10}$$

where

K_s = equipment shaft stiffness (lb/in.)
K_B = equipment bearing stiffness (lb/in.)

Finally, the maximum speed based on the critical speed becomes

$$N_c = \frac{1}{2\pi\sqrt{\dfrac{K_e g}{W}}} \tag{3.11}$$

$$N_s = \frac{N_c}{\text{F.S.}} \tag{3.12}$$

where

F.S. = factor of safety (ranges from 1.4 to 2.0, usually 1.5) *c. Other Considerations.* Many other considerations may affect a coupling's actual maximum speed capabilities, including

1. Heat generated from windage, flexing, and/or sliding and rolling of mating parts
2. Limitations on lubricants used (e.g., separation of grease)
3. Forces generated from unbalance
4. System characteristics

9. Ratings in Summary

The important thing to remember is that the inverse of a service factor or factor of safety is an ignorance factor. What this means is that when little is known about the operating spectrum, a large service factor should be applied. When the operating spectrum is known in detail, the service factor can be reduced. Similarly, if the properties of a coupling material are not exactly known or the method of calculating the stresses is not precise, a large factor of safety should be used. However, when the material properties and the method of calculating the stresses are known precisely, a small factor of safety can be used.

Sometimes the application of service and safety factors to situations in which the operating spectrum, material properties, and stresses are known precisely can cause the use of unnecessarily large and heavy couplings. Similarly, not knowing the operating spectrum, material properties, or stresses and not applying the appropriate service factor or factor of safety can lead to the use of undersized couplings. Ultimately, this can result in coupling failure or even equipment failure.

B. Coupling Component Strength Limits

1. Allowable Limits

Depending on the type of loading, the basis for establishing allowable stress limits may vary. Usually, loads can be classified into three groups:

1. Normal or steady-state
2. Cyclic or reversing
3. Infrequent or peak

Therefore, for comparison purposes we can conservatively establish three approaches to allowable stress limits.

a. Normal Steady-State Loads

$$S_t = \frac{S_{\text{yld}}}{\text{F.S.}} \tag{3.13}$$

$$\tau = \frac{0.577 S_{\text{yld}}}{\text{F.S.}} \tag{3.14}$$

where

S_t = allowable tensile stress (psi)
τ = allowable shear stress (psi)
S_{yld} = yield strength (psi)
F.S. = factor of safety (1.25–2)

b. Cyclic or Reversing Loads. Where loading produces primarily tensile or shear stresses,

$$S_{te} = \frac{S_{\text{end}}}{\text{F.S.} \times K} \tag{3.15}$$

$$\tau_e = \frac{0.577 S_{\text{end}}}{\text{F.S.} \times K} \tag{3.16}$$

where

S_{te} = allowable endurance limit of tensile stress (psi)
S_{end} = tensile endurance limit (psi) = $0.5 S_{\text{ult}}$
K = stress concentration factor
τ_e = allowable endurance limit of shear stress (psi)

Factors of safety are 1.25–2, usually 1.5. K usually ranges from 1.25 to 2.

Where combined tensile and shear stresses are present,

$$S_D \text{ or } S_M = \frac{S_t}{2} + \sqrt{\left(\frac{S_t}{2}\right)^2 + \tau^2} \tag{3.17}$$

$$\frac{1}{\text{F.S.}} = \frac{S_D}{S_{\text{end}}} + \frac{S_M}{S_{\text{ult}}} \tag{3.18}$$

where

S_D = combined dynamic stresses (psi)
S_M = combined steady-state stresses (psi)

S_{ult} = ultimate strength (psi)

Factors of safety are 1.5–2.

c. Infrequent or Peak Loads. In these cases F.S. ranges from 1.0 to 1.25, typically 1.1.

$$S_t = \frac{S_{\mathrm{yld}}}{\mathrm{F.S.}} \tag{3.19}$$

$$\tau = \frac{0.577 S_{\mathrm{yld}}}{\mathrm{F.S.}} \tag{3.20}$$

2. Typical Material Properties of Material for Couplings

Figure 3.6 shows the properties of typical materials used for various components of couplings. The following are listed: material hardness, tensile ultimate strength, tensile yield strength, shear ultimate strength, and shear yield strength.

Material	Hardness (bhn)	Tensile ultimate strength, S_{ult} (psi)	Tensile yield strength, S_{yld} (psi)	Shear ultimate strength, τ_{ult} (psi)	Shear ultimate strength, τ_{yld} (psi)
Aluminum	50	25,000	12,000	18,800	6,900
Die casting	75	50,000	25,000	37,500	14,400
Aluminum					
bar-forging,	100	60,000	35,000	45,000	20,200
heat treated	150	80,000	70,000	60,000	40,400
Carbon	110	50,000	30,000	37,500	17,300
	160	85,000	50,000	63,800	28,900
	250	110,000	70,000	82,500	40,400
Alloy steel	300	135,000	110,000	101,300	63,500
	330	150,000	120,000	112,500	69,200
	360	160,000	135,000	120,000	77,900
Surface-	450	200,000	170,000	150,000	98,100
hardened	550	250,000	210,000	187,500	121,200
alloy steel	600	275,000	235,000	206,300	135,600

$\tau_{\mathrm{ult}} = 0.75 \times S_{\mathrm{ult}}$; $\tau_{\mathrm{yld}} = 0.577 \times S_{\mathrm{ult}}$

FIGURE 3.6 Properties of coupling components.

bhn	Normal psi	Maximum psi
110	32,500	65,000
160	45,000	90,000
250	65,000	130,000
300	72,500	145,000
330	80,000	160,000
360	90,000	180,000
450	125,000	250,000
550	137,500	275,000
600	150,000	300,000

FIGURE 3.7 Compressive stress limits of steel.

3. Compressive Stress Limits for Various Steels

Figure 3.7 shows compressive stress limits for various hardnesses of steels. These limits are strongly dependent on whether relative motion takes place, on the type of lubricant used, and on whether lubrication is present. The limits given in this table are typical limits for nonsliding components such as, keys and splines. For sliding parts the allowable values would be approximately one-fourth to one-third of the given values. It should also be noted that in some applications with good lubricants and highly wear resistant materials (nitrided or carburized) for parts that roll and/or slide (e.g., high-angle low-speed gear couplings), the values in the table can be approached.

4. Allowable Stress Limits for Various Coupling Materials

Figure 3.8 shows typical allowable limits for various coupling materials using the following values for F.S. and *K*.

Steady-state normal conditions:

Tensile allowable F.S. = 1.5
Shear allowable F.S. = 1.5

Cyclic-reversing conditions:

Tensile: $K = 1.5$; F.S. = 1.35
Shear: $K = 1.3$; F.S. = 1.35

		Steady-state allowables		Cyclic-reversing allowables		Peak allowables	
Material	Hardness (bhn)	S_t (psi)	τ (psi)	S_t (psi)	τ (psi)	S_t (psi)	τ (psi)
Aluminum	50	8,000	4,600	6,200	4,100	10,900	6,300
Die casting	75	16,700	9,600	12,300	8,200	22,700	13,100
Aluminum							
bar-forging,	100	23,300	13,500	14,800	9,900	31,800	18,400
heat treated	150	46,700	26,900	19,800	13,200	63,600	36,700
Carbon	110	20,000	11,500	12,300	8,200	27,300	15,700
	160	33,300	19,300	21,000	14,000	45,500	26,300
	250	46,700	26,900	27,200	18,100	63,600	36,700
Alloy steel	300	73,300	42,300	33,300	22,200	100,000	57,700
	330	80,000	46,100	37,000	24,700	109,100	62,900
	360	90,000	51,900	39,500	26,300	122,700	70,800
Surface-	450	113,300	65,400	49,400	32,900	154,500	89,200
hardened	550	140,000	80,800	61,700	41,100	190,900	110,200
alloy steel	600	156,700	90,400	67,900	45,200	213,600	123,300

FIGURE 3.8 Typical allowable limits for various materials for coupling components.

Infrequent peak conditions:

Tensile: F.S. = 1.1
Shear: F.S. = 1.1

C. General Equations

1. Torque

$$\text{Torque } (T) = \text{horsepower (hp)} \times 63{,}000/\text{rpm} = \text{in.-lb} \tag{3.21}$$

$$\text{Torque } (T) = \text{kilowatts (kW)} \times 84{,}420/\text{rpm} = \text{in.-lb} \tag{3.22}$$

$$\text{Torque } (T) = \text{newton-meters (N·m)} \times 8.85 = \text{in.-lb} \tag{3.23}$$

2. Misalignment

a. Angular Misalignment. With the exception of some elastomeric couplings, a single-element flexible coupling can usually accommodate only angular misalignment (α) (see Figure 3.9A). A double-element

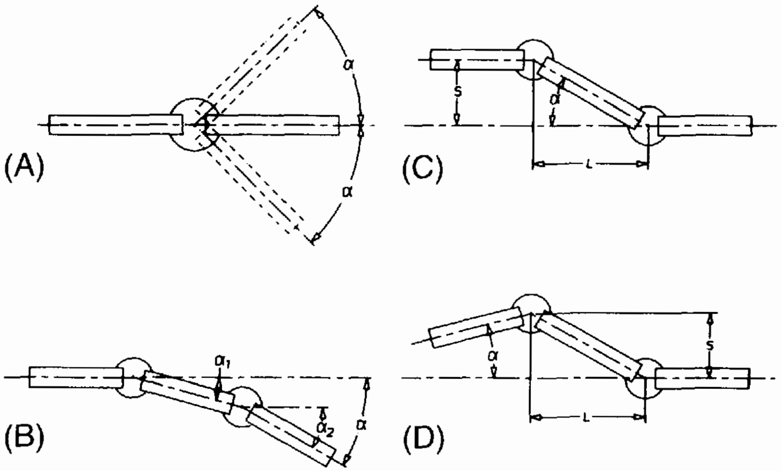


Figure 3.9 Types of shaft misalignments. (A) Angular misalignment for single element. (B) Angular misalignment for double element. (C) Offset misalignment. (D) Offset and angular misalignment combined.

flexible coupling can handle both angular misalignment (see Figure 3.9B) and offset misalignment.

$$\text{Angular misalignment } \alpha = \alpha_1 + \alpha_2 \qquad (3.24)$$

b. *Offset Misalignment.* See Figure 3.9C.

$$S = L \tan \alpha \qquad (3.25)$$

where

S = offset distance
L = length between flex points

α (deg)	Offset in./in. shaft separation
1/4	0.004
1/2	0.008
3/4	0.012
1	0.018
1 1/2	0.026
2	0.035
3	0.052
4	0.070
5	0.087
6	0.100

c. *Combined Offset and Angular Misalignment.* See Figure 3.9D.

$$\alpha_{\text{allowable}} = \alpha_{\text{rated}} - \tan^{-1}\left(\frac{S}{L}\right) \qquad (3.26)$$

3. Weight (*W*), Flywheel Effect (*WR*²), and Torsional Stiffness (*K*)

For a solid disk:

Weight = lb

$$\text{Steel:} \quad W = 0.223 L D_o^2 \qquad (3.27)$$

$$\text{Aluminum:} \quad W = 0.075 L D_o^2 \qquad (3.28)$$

WR^2 = lb-in.^{2} (psi)

$$WR^2 = \frac{WD_o^2}{8} \tag{3.29}$$

where

D_o = outside diameter (in.)
L = length (in.)

Torsional stiffness (K) = in.-lb/rad

$$\text{Steel:} \quad K = 1.13 \times 10^6\, D_o^4 \tag{3.30}$$

$$\text{Aluminum:} \quad K = 0.376 \times 10^6\, D_o^4 \tag{3.31}$$

Tubular (disk with a hole):

Weight = lb

$$\text{Steel:} \quad W = 0.223L(D_o^2 - D_i^2) \tag{3.32}$$

$$\text{Aluminum:} \quad W = 0.075L(D_o^2 - D_i^2) \tag{3.33}$$

WR^2 = lb-in.^{2} (psi)

$$WR = \frac{W}{8}\,(D_o^2 + D_i^2) \tag{3.34}$$

where

D_i = inside diameter (in.)
Torsional stiffness (K) = in.-lb/rad

$$\text{Steel:} \quad K = \frac{1.13 \times 10^6\,(D_o^4 - D_i^4)}{L} \tag{3.35}$$

$$\text{Aluminum:} \quad K = \frac{0.375 \times 10^6\,(D_o^4 - D_i^4)}{L} \tag{3.36}$$

4. Bending Moment

Most couplings exhibit a bending stiffness. If you flex them, they produce a reaction. Bending stiffness (K_B) is usually expressed as

$$K_B = \text{in.-lb/deg} \tag{3.37}$$

Therefore, the moment reaction (M) per flex element is

$$M = K_B\alpha = \text{in.-lb} \tag{3.38}$$

Note: Some couplings (e.g., gear couplings) exhibit bending moments that are functions of many factors (torque, misalignment, load distribution, and coefficient of friction).

5. Axial Force

There are two types of axial force: force from the sliding of mating parts and force from flexing of material.

Sliding force (F) (e.g., gear and slider block couplings):

$$F = \frac{T\mu}{R} = \text{lb} \tag{3.39}$$

where

T = torque (in.-lb)
μ = coefficient of friction
R = radius at which sliding occurs (in.)

Flexing force (F) (e.g., rubber and metallic element couplings):

$$F = K_A S = \text{lb} \tag{3.40}$$

where

K_A = axial stiffness (lb/in. per mesh)
S = axial deflection (in.)

Note: For two meshes, $K_A' = K_A/2$.

D. Component Stress Equations

1. Shaft Stresses

a. Circular Shift. The stress on a circular shaft depends on whether the shaft is solid or tubular.

Solid (Figure 3.10A):

$$\tau_S = \frac{16T}{\pi D_o^3} \tag{3.41}$$

Tubular (Figure 3.10B):

$$\tau_S = \frac{16TD_o}{\pi(D_o^4 - D_i^4)} \tag{3.42}$$

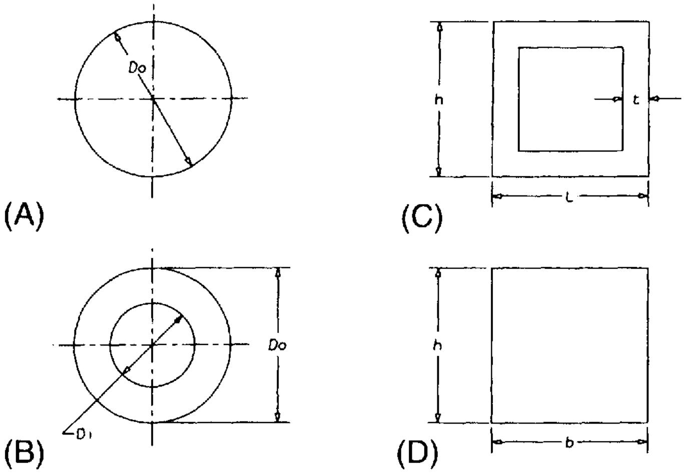


FIGURE 3.10 Types of shaft shapes. (A) Circular shaft, solid. (B) Circular shaft, tubular. (C) Rectangular shaft, tubular. (D) Rectangular shaft, solid.

b. Noncircular Tubing. Rectangular section (Figure 3.10C):

$$\tau_S = \frac{(15b + 9h)T}{5b^2h^2} \tag{3.43}$$

Square solid (Figure 3.10D):

$$b = h$$

$$\tau_S = \frac{24T}{5h^3} \tag{3.44}$$

Any shape:

$$\tau_S = \frac{T}{2At} \tag{3.45}$$

where

A = area enclosed by centerline
t = wall thickness

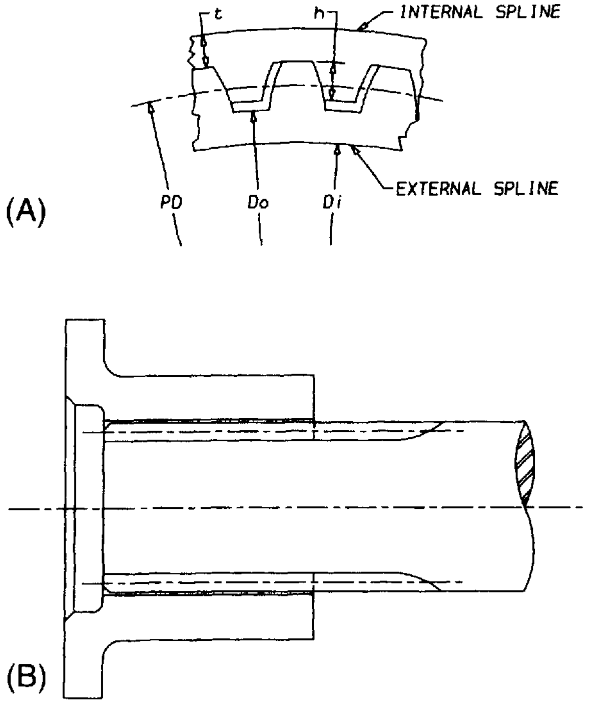


Figure 3.11 Spline dimensions. (A) Spline teeth dimensions. (B) Spline hub dimensions.

2. Spline Stresses

For spline stresses, see Figure 3.11 (*Note:* Stresses assume that 100% of the teeth carry the load.)

Compressive stress:

$$S_c = \frac{2T}{n \times \text{P.D.} \times L \times h} \tag{3.46}$$

Shear stress:

$$S_S = \frac{2T}{\text{P.D.} \times L \times n \times t_c} \tag{3.47}$$

Bending stress:

$$S_b = \frac{2T}{(\text{P.D.})^2 L \times Y} \tag{3.48}$$

Bursting stress (hub):

$$S_t = \frac{T}{\pi \times \text{P.D.} \times L}\left(\frac{\tan\theta}{t}\right) \tag{3.49}$$

Shaft shear stress (shaft):

$$S_S = \frac{16T \times D_o}{\pi(D_o^4 - D_i^4)} \tag{3.50}$$

where

T = torque
n = number of teeth
P.D. = pitch diameter
L = effective face width (in.)
h = height of spline tooth
θ = pressure angle
t = hub wall thickness
D_o = diameter at root of shaft tooth
D_i = inside diameter (I.D.) of shaft (zero for solid shaft)
$t_c \simeq$ P.D./2n
$Y \simeq$ 1.5 for 30° pressure angle splines

3. Key Stresses

Square or rectangular keys (Figure 3.12):

$$T = \frac{FDn}{2} \tag{3.51}$$

$$F = \frac{2T}{Dn} \tag{3.52}$$

$$S_s = \text{shear stress} = \frac{F}{wL} = \frac{2T}{WLDn} \tag{3.53}$$

$$S_c = \text{compressive stress} = \frac{2F}{hL} = \frac{4T}{hLDn} \tag{3.54}$$

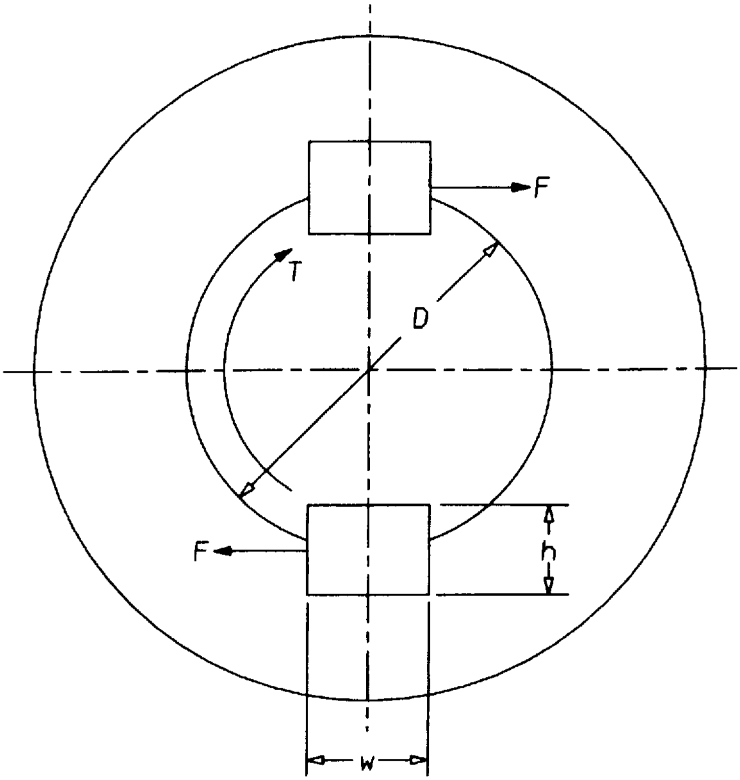


FIGURE 3.12 Square or rectangular key loading and dimensions.

where

L = length of key (in.)
n = number of keys
D = shaft diameter (in.)
w = key width (in.)
h = key height (in.)
T = torque (in.-lb)
F = force (in.-lb)

For allowable key stress limits see Figure 3.13.

	Shear τ, normal psi	Compressive S_C, normal psi	Shear τ, peak psi	Compressive S_C, peak psi
Commercial key stocks	12,500	32,500	23,000	65,000
1045 (170 bhn)	15,000	45,000	27,500	90,000
4140–4130 HT (270 bhn)	30,000	70,000	55,000	140,000

FIGURE 3.13 Key limits.

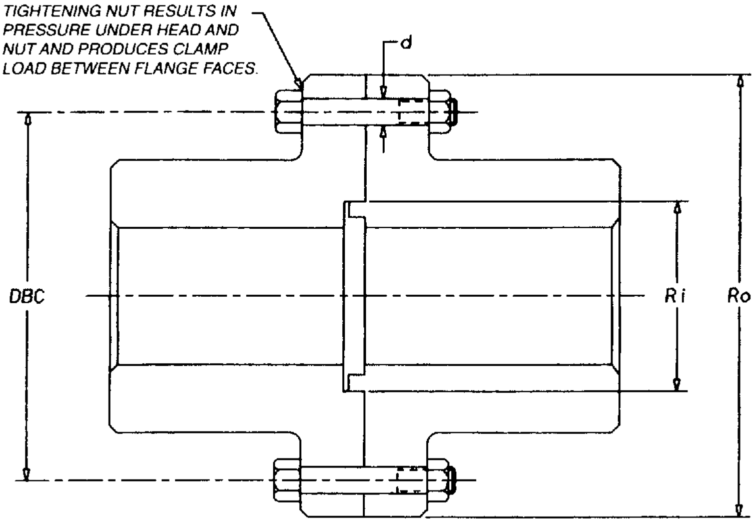

TIGHTENING NUT RESULTS IN PRESSURE UNDER HEAD AND NUT AND PRODUCES CLAMP LOAD BETWEEN FLANGE FACES.
d
DBC
Ri
Ro

FIGURE 3.14 Dimensions for bolted flange connection.

E. Flange Connections

See Figure 3.14.

1. Friction Capacity

$$T = F\mu R_f \tag{3.55}$$

F = clamp load from bolts

$$F = \frac{T_B}{\mu_B \mathrm{d}}\, n \tag{3.56}$$

where

T_B = tightening torque of bolt (in.-lb)
n = number of bolts
d = diameter of bolt (in.)
μ_B = coefficient of friction at bolt threads (0.2 for dry threads)
μ = coefficient of friction at flange faces (0.1–0.2, usually 0.15)
R_f = friction radius (in.)

$$R_f = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right) \tag{3.57}$$

where

R_o = outside radius of clamp (in.)
R_i = inside radius of clamp (in.)

2. Bolt Stresses for Flange Couplings

See Figure 3.14.

a. *Shear Stress.* Allowables shear stress values are

$$\text{At normal operation, } \tau_A = \frac{0.30 S_{\text{ult}}}{1.5} = 0.20 S_{\text{ult}} \tag{3.58}$$

$$\text{At peak operation, } \tau_A = 0.55 S_{\text{yld}} \tag{3.59}$$

where

S_{ult} = ultimate strength of material (psi)
S_{yld} = yield strength of material (psi)

To determine shear stress, the equation to use is

$$\tau_S = \frac{8T}{(\text{D.B.C.})(\%n)(\pi d)} \tag{3.60}$$

where

T = operating torque (in.-lb)
D.B.C. = diameter of the bolt circle (in.)
d = bolt body diameter

n = number of bolts

$\%n$ = percentage of bolts loaded: fitted bolts 0.000–0.006 loose, 100%; clearance bolts 0.006–0.032 loose, 75%

For very loose bolts with over 0.032 clearance or very long bolts of approximately 10*d*, the analysis above may not be applicable. For very loose and long bolts, the bending stresses produced in a bolt should be considered.

b. Tensile Stress. Allowable value is

$$S_t = 0.8S_{\text{yld}} \tag{3.61}$$

where

S_t = stress from tightening bolt

$$S_t = \frac{F_n}{A_S} \tag{3.62}$$

where

A_S = bolt thread root stress area (in.)

F_n = clamp load produced by bolt (lb)

$$S_t = 0.25\text{–}0.75\ S_{\text{yld}} \tag{3.63}$$

S_t is usually 0.5.

c. Bolt Tightening Torque (T_B). From the foregoing we can determine the tightening torque T_B:

$$F_n = 0.5S_{\text{yld}}\,A_S = \text{lb} \tag{3.64}$$

$$T_B = F_n\mu d = \text{in.–lb} \tag{3.65}$$

$$T_B = 0.2F_n\,d = \text{in.–lb} \tag{3.66}$$

where

μ = coefficient of friction at threads

For different lubrication conditions C of the threads, use Figure 3.15 to determine the value T_B.

$$T_B' = T_B \times C \tag{3.67}$$

Condition factor	Condition	Coefficient of friction of thread, μ
1	Dry threads—unplated or black-oxided bolts	0.2
0.75%	Lube threads—unplated or black-oxided bolts	0.15
0.75%	Dry threads—cadmium-plated bolts	0.15
0.56	Lube threads—cadmium-plated bolts	0.11

FIGURE 3.15 Frictional factors for various bolt thread conditions.

Figure 3.16 shows properties of the four most common grades of bolts used with couplings.

d. Combined Bolt Stress for Flange Connections

$$S_A \text{ (allowable)} = 0.80 S_{\text{yld}} \tag{3.68}$$

$$S_T = \frac{S_t}{2} + \sqrt{\left(\frac{S_t}{2}\right)^2 + \tau_s^2} \tag{3.69}$$

3. Flange Stresses

See Figure 3.17. Most flexible couplings have flanges (rigids, spacers, etc.) that are loaded, usually with moments and also in shear. Generally, the stresses at point A can be approximated by the following equations when $t \simeq w$.

Bending stress (S_B):

$$S_B = \frac{MC}{I} \times K \tag{3.70}$$

where

K = stress concentration (usually 3 or less)

M = applied moments (any external moment or the moment gen-

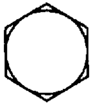
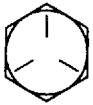
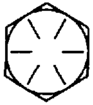
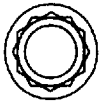
S.A.E. GRADE	HEADSTYLE	BOLT SIZE DIA. IN.	PROOF LOAD	TENSILE STRENGTH	HARDNESS ROCKWELL	MATERIAL - HEAT TREATMENT
2		Up to 1/2 incl. Over 1/2 to 3/4 Over 3/4 to 1-1/2	55,000 52,000 28,000	69,000 64,000 65,000	100 B Max. 100 B Max. 95 B Max.	Low or medium carbon steel: C 0.28 max., and S 0.05 max. For bolts over 6 in. long, or over 3/4 in. dia., carbon may be as high as 0.55.
5		Up to 3/4 incl. Over 3/4 to 1 Over 1 to 1-1/2	85,000 78,000 74,000	120,000 115,000 105,000	23-32 C 27-32 C 19-30 C	Medium carbon steel: C 0.28 to 0.55 P 0.04 max., and S 0.05 max., Quenched and tempered at a minimum temperature of 800°F.
8		Up to 1-1/2 incl.	120,000	150,000	32-38 C	Medium carbon fine grain alloy steel (b): C 0.28 to 0.55. p 0.04 max. and S 0.05 max., providing sufficient hardenability to have a minimum oil quenched hardness of 47 RC at the center of the threaded section one diameter from the end of the bolt. Oil quenched and tempered at a minimum temperature of 800°F.
12-point Aircraft Style		Up to 1-1/4	144,000	160,000	34-40 C	Alloy steel usually 4140, 4340, and others.

FIGURE 3.16 Properties of common grades of coupling bolts.

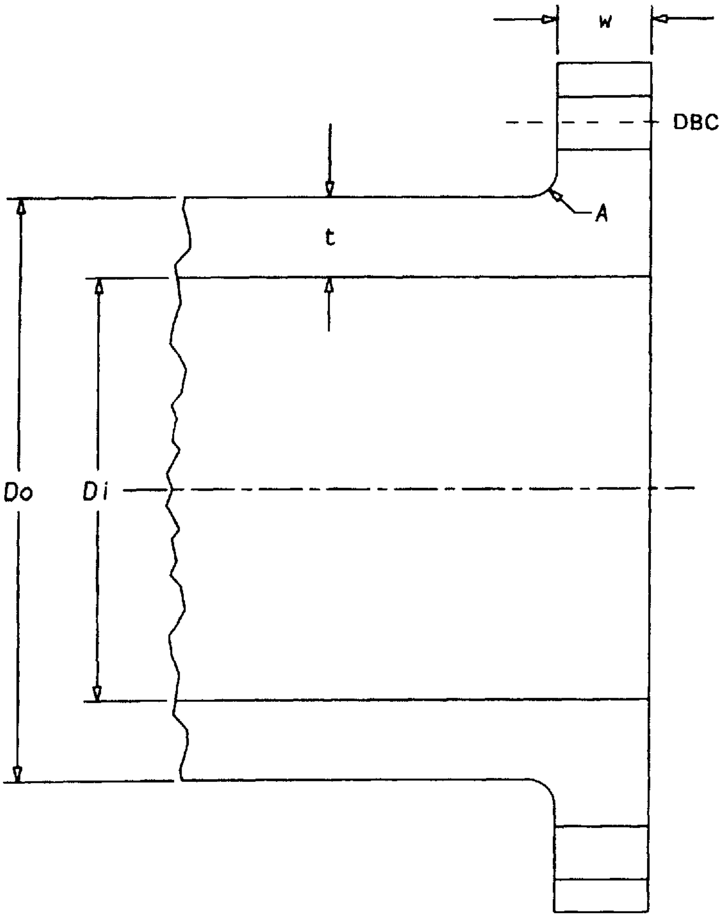


FIGURE 3.17 Flange dimensions for calculating flange stresses.

erated by the flexing element of the coupling (in.-lb) (see Section 11.C.-4)

$$I = 0.049\,(D_o^4 - D_i^4) \tag{3.71}$$

and

$$C = \frac{D_o + D_i}{2}$$

Shear stress (τ):

$$\tau = \frac{16TD_oK}{\pi(D_o^4 - D_i^4)} \tag{3.72}$$

Combined stress:

$$S_M \text{ (steady-state stress)} = \frac{S_t}{2} + \sqrt{\left(\frac{S_t}{2}\right)^2 + \tau^2} \tag{3.73}$$

where S_t is any steady-state tensile stress [e.g., centrifugal stress $S_t = \rho V^2/g$ or any other axial loading on the flange (external or internal)].

$$\frac{1}{\text{F.S.}} = \frac{S_M}{S_{\text{ult}}} + \frac{S_B}{S_{\text{end}}} \tag{3.74}$$

where

F.S. = factor of safety
S_{ult} = ultimate strength of material (psi)

F. Hub Stresses and Capacities

1. Hub Bursting Stress for Hubs with Keyways

Figure 3.18. The following approach is applicable when the interference is less than 0.0005 in./in. shaft diameter and X is greater than $(D_o - D_i)/8$.

Stress:

$$S_t = \frac{T}{AYn} \tag{3.75}$$

where

n = number of keys
Y = radius of applied force (in.)
A = area = XL [L = length of hub (in.)]

$$X = \sqrt{\left(\frac{D_o}{2}\right)^2 - \left(\frac{W}{2}\right)^2} - \frac{h}{2} - \sqrt{\left(\frac{D_i}{2}\right)^2 - \left(\frac{w}{2}\right)^2} \tag{3.76}$$

$$Y = \frac{2D_i + h}{4} \tag{3.77}$$

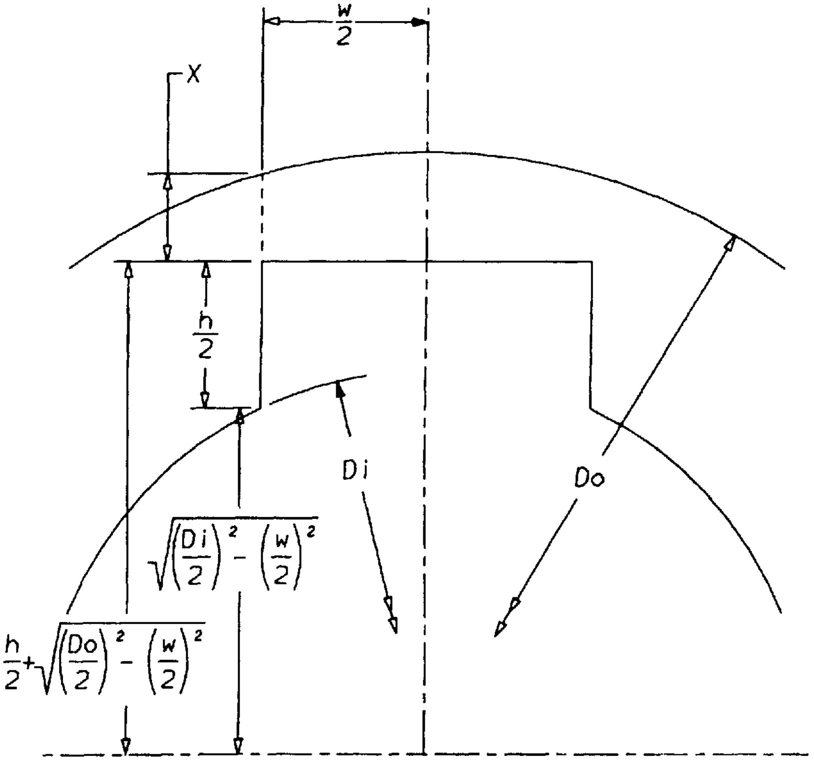

$\frac{w}{2}$
X
$\frac{h}{2}$
Di
Do
$\sqrt{\left(\frac{Di}{2}\right)^2 - \left(\frac{w}{2}\right)^2}$
$\frac{h}{2}+\sqrt{\left(\frac{Do}{2}\right)^2 - \left(\frac{w}{2}\right)^2}$

FIGURE 3.18 Dimensions for calculating hub bursting stress.

where

D_o = outside diameter
D_i = inside diameter
h = height of key
w = width of key

2. Hub Stress and Capacity for Keyless Fit Hubs or Shrink Drives

See Figure 3.19. The design of keyless fit hubs for couplings requires the analysis of three types of factors:

1. The amount of interference required to handle the required torque
2. The pressure required to mount or dismount the hub
3. The stresses in the hub during mounting or dismounting and during operation

The equations given in this section for pressure and stress are for hubs of relatively uniform cross section. When a hub has multiple outside or inside diameters (stepped bores) as shown in Figure 3.20, the maximum stress is at the section with the smallest diameter and total torque capacity is the sum of the separate section capacities.

Shrink or interference fit calculations are based on Lame's equations for a thick-walled cylinder under internal pressure.

The bore used in the calculations of pressure for tapered and cylindrical bores is shown in Figures 3.19A and 3.19B. The analysis is based on the assumption that the shaft and hub have the same finish, hardness, and modulus of elasticity.

a. Torque Capacity of a Shrink Fit. The interference required is dependent on the torque to be transmitted, the coefficient of friction used, the dimensions of the hub, and the operating speed. The interference range is determined by either the minimum/maximum shaft and bore diameters for cylindrical bores or by the minimum/maximum hub

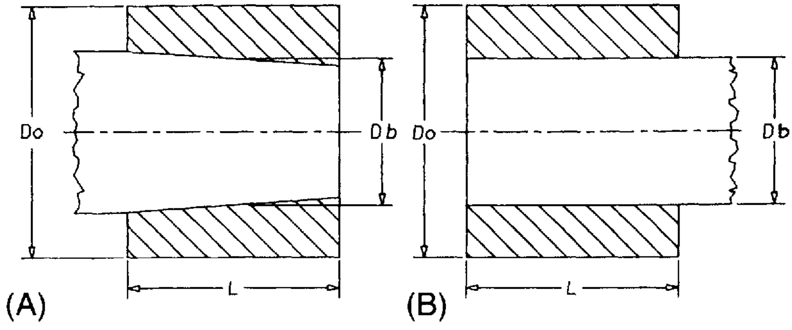


FIGURE 3.19 Dimensions for keyless hubs. (A) Taper shaft. (B) Straight shaft.

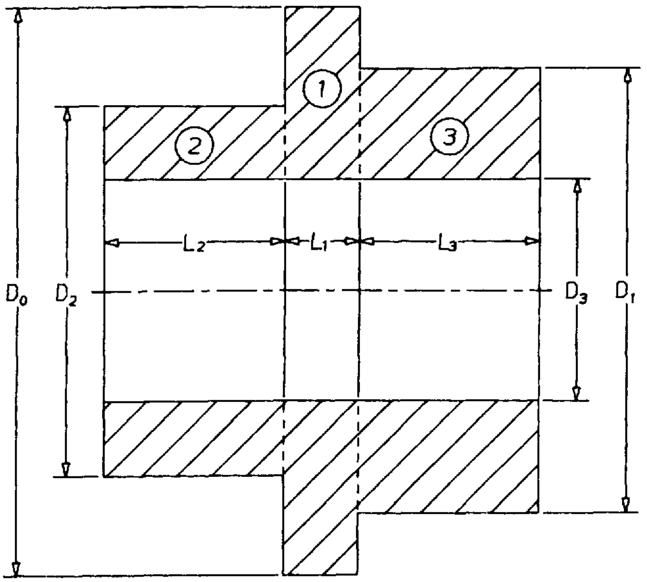


FIGURE 3.20 Dimensions for determining torque capacity of stepped or flanged hubs.

advance for tapered bores. Torque capacity for hubs with steps or flanges (Figure 3.20) requires breaking the hub into sections and then summing the section values.

b. Apparent Coefficient of Friction. The apparent coefficient of friction, μ, between the hub and the shaft can vary widely depending on such factors as cleanliness, oil type, contact pattern, surface finish, and rate of load application. Values that are commonly used are

$\mu = 0.12–0.15$ if hydraulically mounted with each surface having a finish of 32 μin., rms maximum, and adequate draining time after mounting.

$\mu = 0.15–0.20$ if heat-shrink-mounted with clean and dry surfaces, each having a surface finish of 32 μin., rms maximum.

c. Torque Capacity (T) of Hub-to-Shaft Juncture. The torque capacity T of the hub-to-shaft juncture is a function of the hub dimensions, the interference, the hub and shaft materials, centrifugal expansion, and the coefficient of friction. The capacity should exceed the momentary torque (short-circuit torque) for the application. When the momentary torque is well defined, the hub-to-shaft torque capacity should exceed the momentary torque by a minimum factor of 1.15.

d. Minimum Pressure at the Hub Bore (p_{min}). The minimum pressure (p_{min}) at the hub bore is based on the design torque capacity, the apparent coefficient of friction, and the dimensions of the hub. The minimum pressure is calculated for the smallest hub section, which results in the lowest torque capacity of the hub-to-shaft juncture. It is calculated as follows:

$$p_{\text{min}} = \frac{2T}{\pi D_b^2 L \mu} \tag{3.78}$$

where

T = design torque capacity of hub-to-shaftjuncture, in.-lb
D_b = average bore diameter of section, in.
L = length of hub engagement, in.
μ = apparent coefficient of friction

e. Minimum Interference Fit Required to Handle the Design Torque and the Operating Speed (I_{min}). The minimum interference fit, I_{min}, required to transmit the design torque capacity at the operating speed is based on the minimum pressure as calculated plus the loss of interference due to centrifugal expansion of the hub due to rotation, I_c. It is calculated for the smallest hub section, which results in a conservative torque capacity of the hub-to-shaft juncture. It is calculated as follows:

$$I_{\text{min}} = \frac{2p_{\text{min}} D_b}{E(1 - C_e^2)} + I_c \tag{3.79}$$

where

E = modulus of elasticity, lb/in^2.
C_e = ratio of the average bore diameter of the section to the outside diameter of the section being analyzed, D_b/D_o

$$I_c = 5.52 \times 10^{-14} N^2 D_o^2 D_b \tag{3.80}$$

f. Pressure Required to Mount or Dismount the Hub. The calculated pressure to mount or dismount the hub is based on the pressure generated at the hub bore from the maximum interference and a uniform hub section. The actual pressure required to mount or dismount the hub is somewhat greater than the calculated value.

g. Calculated Pressure to Mount or Dismount the Hub Due to Maximum Interference Fit (p_l). The calculated pressure, p_1, to mount or dismount the hub is based on the maximum interference. The maximum interference fit is equal to the minimum interference as calculated plus the tolerances of the shaft and the hub bore for cylindrical shafts or the fit increase due to the advance tolerance in the case of tapered bored hubs. The pressure is calculated as follows:

$$p_1 = \frac{EI_{\max}}{2D_h}(1 - C_e^2) \qquad (3.81)$$

h. Pressure at the Bore Required to Mount or Dismount the Hub (p_2). The maximum recommended pressure, p_2, at the bore required to mount or dismount the hub, is based on the pressure, p_1, required to expand the hub section.

$$p_2 = 1.1\, p_1 \qquad (3.82)$$

The 1.1 multiplier is based on industry practice to account for the increased pressure required for dismounting a hub of uniform cross section. For hubs of nonuniform cross section, a more rigorous analysis is required that is beyond the scope of this standard.

i. Hub Stress While Mounting or Dismounting (S_{max}). The maximum hub stress, $S_{\max}$, due to hydraulic pressure when mounting or dismounting is based on the distortion energy (von Mises) criterion.

$$S_{\max} = p_2 \frac{\sqrt{(3 + C_e^4)}}{1 - C_e^2} \qquad (3.83)$$

where C_e is calculated for the thinnest hub section; i.e., maximum C_e.

Industry practice is to limit the maximum stress due to mounting or dismounting to 90% of the yield strength.

j. Hub Stress While Rotating (S_{rot}). The combined stress, S_{rot}, in the hub while rotating is calculated by the distortion energy (von Mises) criterion for each hub section as follows:

$$S_{rot} = \sqrt{[(S_H + S_V)^2 + (S_R + S_{RV})^2 - (S_H + S_V)(S_R + S_{RV})]} \tag{3.84}$$

where

S_H = maximum hoop stress in the mounted hub including the loss due to rotation, lb/in^2.

S_V = hoop stress in the hub due to rotational speed, lb/in^2.

S_R = radial stress in the hub bore due to interference fit of the mounted hub while rotating, lb/in^2.

S_{RV} = radial stress in the hub due to rotational speed, lb/in^2.

$$S_H = \frac{E(I_{\max} - I_C)(1 + C_e^2)}{2D_b} \tag{3.85}$$

$$I_C = 5.52 \times 10^{-14} N^2 D_o^2 D_b \tag{3.86}$$

$$S_V = 0.502 \left(\frac{N}{1000}\right)^2 (3.3 D_o^2 + 0.7 D_b^2) \tag{3.87}$$

$$S_R = \frac{-E(I_{\max} - I_C)(1 - C_e^2)}{2D_b} \tag{3.88}$$

$$S_{RV} = 0 \tag{3.89}$$

Note: $S_{rot} \leq 0.7\, S_y$, based on industry practice, which limits the stress to 70% of the yield strength of the material used.

Rotational stresses S_v and S_{rv} are based on a homogeneous circular disk of uniform thickness with a center hole.

k. Hub Bore Surface Finish. The hub bore surface finish should be 32 μin. rms or better. It is recommended that the shaft have a similar finish. It is also recommended that the surface be a ground finish rather than turned.

III. BALANCING OF COUPLINGS

The subject of coupling balance (or unbalance) is a topic often discussed wherever the subject of equipment vibration comes up. Often a coupling is blamed for shaking equipment when it is actually the equipment shaking the coupling. Therefore to keep equipment from vibrating we will just balance the "H" out of the coupling. This philosophy can be costly and unnecessary. One really needs to understand what affects the balance of a coupling and how these effects contribute to unbalance and cause the equipment to vibrate.

There are in the industry four balance standards that are most often applied to couplings: API 671, AGMA 9002, ANSI S2 19-1989, and ISO 1940/1 (First edition, 1986-09-01). Of all these only one was specifically written for couplings, AGMA 9002. The other three use tolerances that were developed for rotors or other rotating parts.

This section will explain what coupling balance level is really needed for various types of equipment, how coupling unbalance affects the vibration of equipment, how the arbitrary balance criteria should be applied, the difference between residual balance limit and potential unbalance limits, and how to improve a coupling's balance. Also discussed will be what balance levels can be achieved for various types of couplings.

A. Balancing Fundamentals

1. Units for Balance or Unbalance

The unbalance in a piece of rotating equipment is usually expressed in terms of unbalance weight (ounces or grams) and its distance from the rotating centerline (inches or millimeters). Thus, the unbalance (U) in a part is expressed in ounce-inches (oz-in.) or gram-millimeters (g.mm).

EXAMPLE

An unbalance weight of 4 oz is at a distance of 2 in. from the rotating centerline (Figure 3.21). The unbalance (U) is therefore 8 oz-in. For couplings, it is best to express unbalance in terms of mass displacement. This usually helps bring balance limits into perspective with the tolerances and fits of a coupling and the "real world."

Mass displacement is usually specified as microinches (μin.; 1000 μin. = 1 mil = 0.001 in.), mils or inches, or in metric terms as micrometers (μm) or millimeters (mm) (1000 μm = 1 mm). If a 50 lb (800 oz) part is displaced 0.001 in. (1 mil or 1000 μin.) from the centerline

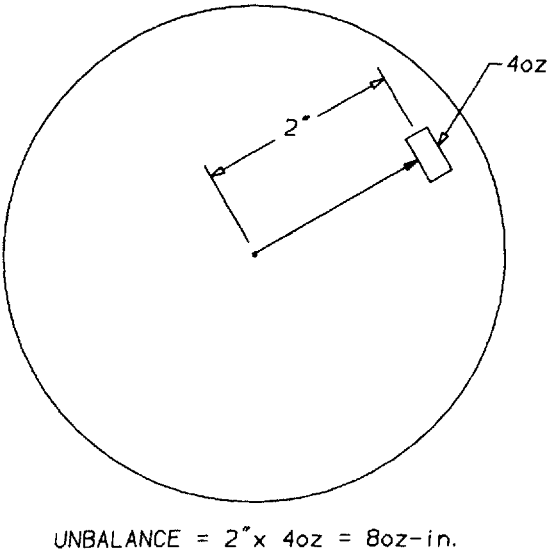


FIGURE 3.21 Unbalance units.

(Figure 3.22), then an 8 oz-in. unbalance (800 oz × 0.001 in.) will occur.

2. Balancing Types

Balancing is a procedure by which the mass distribution of a rotating part is checked and, if required, adjusted (usually by removing metal) so the unbalance force or the vibration of the equipment is reduced.

a. Single-Plane Unbalance. Single-plane unbalance (sometimes called static balance) exists when the center of gravity of a rotating part does not lie on the axis of rotation (Figure 3.23). This part will not be in static equilibrium when placed at random positions about its axis. Single-plane balancing can be done without rotation [much like the static balancing (bubble balancing) that has been done for years on automobile wheels], but most couplings are balanced on a balancing machine and spun (spin balancing is also done today on automobile tires), which is more accurate. Single-plane balancing is usually limited to short

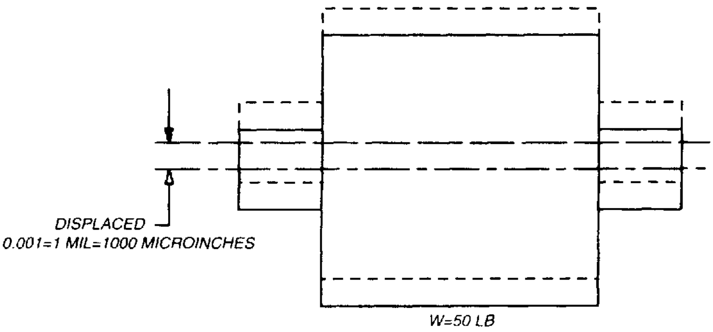


FIGURE 3.22 Displaced mass.

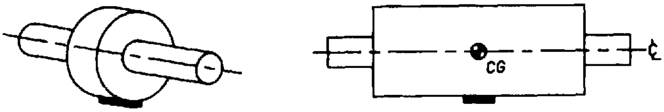


FIGURE 3.23 Single-plane unbalance.

parts, usually with a length between $0.5D_0$ and $1.5D_0$ (depending on the part configuration).

b. Two-plane Unbalance. Two-plane unbalance (sometimes called moment, coupled, or dynamic balance) is present when the unbalance existing in two planes is out of phase as shown in Figure 3.24, but not necessarily 180° out of phase. The principal inertia axis of rotation and the geometric axis are skewed with respect to each other. If not restrained by bearings, a rotating part with this type of unbalance will tend to rotate about its principal inertia axis closest to its geometric axis.

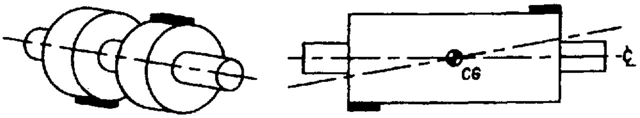


FIGURE 3.24 Two-plane unbalance.

If the moments are equal and opposite, they are referred to as a couple unbalance.

c. Rigid Rotor. A rotor is considered rigid when it can be corrected in any two planes and, after the part is balanced, its unbalance does not significantly exceed the unbalance tolerance limit at any other speed up to the maximum operating speed.

Note: A flexible coupling is an assembly of several components that may have diametral clearances and eccentricities between pilot surfaces of its components. Therefore, it is not appropriate to apply standards and requirements that are written to apply to rotors (without some interpretation).

B. Balancing Definitions

Aligning surface. The axis from which a part is located for the purpose of balancing. See also *Indicating surface.*

Amount of unbalance. The measure of unbalance in a part (or a specific plane), without relationship to its angular position.

Axis of rotation. A line about which a part rotates as determined by journals, fits, or other locating surfaces.

Balance limit tolerance. The potential unbalance specifies a maximum amount of unbalance that might exist in a coupling after balancing (if corrected), disassembly, and reassembly.

Balancing bushing. A bushing or bushing adapter used to mount a coupling component or a coupling assembly onto a balancing mandrel.

Balancing mandrel. A shaft on which coupling components, subassemblies, or the assembled coupling is mounted for balancing purposes. (See Figure 3.25.)

Indicating surface. The axis about which a part is aligned for the purpose of balancing. See also *Aligning surface.*

Mandrel assembly. An assembly that consists of one or more bushings used to support the part mounted on a mandrel (see Figure 3.25).

Mounting fixtures. The tooling that adapts to the balancing machine and provides a surface on which a component or a coupling assembly is mounted.

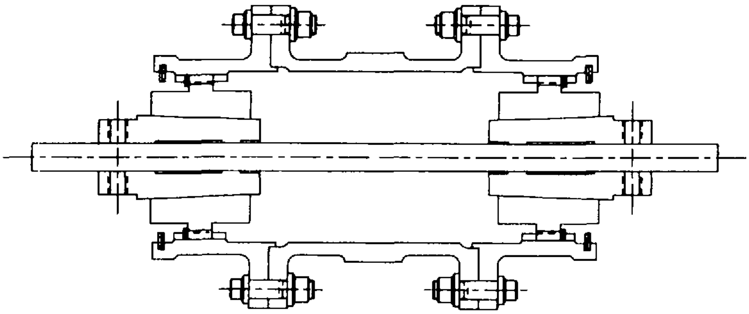


FIGURE 3.25 Mandrel assembly for balancing a gear coupling.

Mounting surface. The surface of a mandrel, bushing, or mandrel assembly on which another part of the balancing tooling, a coupling component, subassembly, or the assembly is mounted. This surface determines the rotational axis of the part being balanced.

Pilot surface. The support surface of a coupling component, subassembly, or assembly upon which another coupling component is mounted or located. Examples are given in Figure 3.26.

Components	Usual pilot surfaces
Rigid hub	Bore, rabbet diameter, bolt circle
Flex hub (gear type)	Bore, hub body O.D., tooth tip diameter
Flanged sleeve (gear type)	Tooth root diameter, end ring I.D., rabbet diameter, bolt circle
Flanged adapter	Rabbet diameter or bolt circle
Flanged stub end adapter	Stub end (shaft) diameter, rabbet diameter, bolt circle
Spool spacer (gear type)	Tooth tip diameter
Flanged spacer	Rabbet diameter or bolt circle
Ring spacer	Rabbet diameter or bolt circle

FIGURE 3.26 Types of coupling pilot surfaces.

Potential unbalance. The maximum amount of unbalance that might exist in a coupling assembly after balancing (if corrected), disassembly, and reassembly.

Principal inertia axis displacement. The movement of the inertia axis that is closest to the axis rotation with respect to the axis of rotation. In some special cases, these two axes may be parallel. In most cases, they are not parallel and are therefore at different distances from each other in the two usual balancing planes.

Residual unbalance. The amount of unbalance left after a part has been balanced. It is equal to or less than the balance limit tolerance for the part, subassembly, or assembly. *Note:* As a check to determine whether a part is balanced, remove the part from the balancing machine and then replace it. The residual unbalance will not necessarily be the same as the original reading. This is due to the potential differences in mounting and/or indicating surface runouts.

C. Coupling Balance

1. Why Balance a Coupling?

One important reason for balancing a coupling is that the forces created by unbalance could be detrimental to the equipment, bearings, and support structures. The amount of force generated by an unbalance is

$$F = 1.77 \times \left(\frac{\text{rpm}}{1000}\right)^2 \times (\text{oz-in.}) = \text{lb} \tag{3.90}$$

$$F = 1.77 \times \left(\frac{\text{rpm}}{1000}\right)^2 \times (\text{wt} \times 16) \times \frac{\mu\text{in.}}{1{,}000{,}000} = \text{lb} \tag{3.91}$$

As can be seen from these equations, the amount of force generated by unbalance is proportional to the square of the rotational speed.

EXAMPLE

For 2 oz-in. of unbalance at 2000 rpm and at 4000 rpm,

F at 2000 rpm = 14.1 lb

F at 4000 rpm = 56.6 lb

This shows that if the speed doubles, the same amount of unbalance produces four times the force.

Another important reason for balancing is to prevent undue vibration.

2. What Causes or Contributes to Unbalance in an Uncorrected Coupling?

a. Inherent Unbalance of Uncorrected Couplings. If the coupling assembly or component is not balanced, an estimate of inherent unbalance caused by manufacturing tolerances can be based on either statistical analysis of the balance data accumulated for couplings manufactured to the same tolerances or calculations of the maximum possible unbalance that could theoretically be produced by the tolerances placed on the parts.

b. Others. Many other factors may contribute to coupling unbalance. The following factors are of principal importance.

Coupling pilot surface eccentricity is any eccentricity of a pilot surface that permits relative radial displacement of the mass axis of mating coupling parts or subassemblies.

Coupling pilot surface clearances are clearances that permit relative radial displacement of the mass axis of the coupling components or subassemblies. *Note:* Some couplings, e.g., gear couplings, must have clearances in order to attain their flexibility.

Hardware unbalance is the unbalance caused by all the coupling hardware, such as fasteners, bolts, nuts, lockwashers, lube plugs, seal rings, gaskets, keys, snap rings, keeper plates, thrust plates, and retainer nuts.

3. What Causes or Contributes to Unbalance of a Balanced Coupling?

Balance tolerance limits are the largest amounts of unbalance (residual) for which no further correction need be made.

Balance fixture or mandrel assembly unbalance is the combined unbalance caused by all components used to balance a coupling, such as mandrel, flanges, adapters, bushings, locking devices, keys, setscrews, nuts, and bolts.

Balance machine errors are mainly due to overall machine sensitivity and error of the driver itself. This unbalance is usually minimal and can usually be ignored.

Mandrel assembly mounting surface eccentricity is the eccentricity with respect to the axis of rotation of the surface of a mandrel assembly upon which the coupling assembly or component is mounted.

Component or assembly indicating surface eccentricity is the eccentricity with respect to the axis of rotation of a surface used to indicate or align a part on a balancing machine. On some parts, this surface may be machined to the axis of rotation rather to the indicating axis.

Coupling pilot surface eccentricity is an eccentricity of a pilot surface that permits relative radial displacement of the mass axis of another coupling part or subassembly during assembly subsequent to the balancing operation. This eccentricity is produced by manufacturing before balancing. For example, most gear couplings that are assembly balanced are balanced with an interference fit between the gear major diameters and after balancing are remachined to provide clearance. This may produce eccentricity.

Coupling pilot surface clearances are the clearances that permit relative displacement of the mass axes of the coupling component or subassemblies on disassembly and reassembly. The radial shift affecting potential unbalance is equal to half of this clearance. *Note:* If this clearance exists in a coupling when it is balanced as an assembly, the potential radial displacement affecting potential unbalance is equal to the full amount of the diametral clearance that exists in the assembly at the time of balance.

Hardware unbalance is the unbalance caused by all the coupling hardware including fasteners, bolts, nuts, lockwashers, lube plugs, seal rings, gaskets, keys, snap rings, keeper plates, thrust plates, retainer nuts, etc.

Many other factors may also contribute to coupling unbalance. The factors mentioned are of principal importance.

4. How to Bring a Coupling into Balance

Couplings can be brought into balance by four basic methods:

a. Tighter manufacturing tolerances
b. Component balancing

c. Assembly balancing
d. Field balancing on the equipment

a. Tighter Manufacturing Tolerances. The majority of unbalance in most couplings comes from the clearance fits that are placed on components so that they can be mass produced and yet be interchangeable. Most couplings are not balanced. The amount of unbalance can be greatly improved by tightening fits and tolerances. If manufacturing tolerances and fits are cut by approximately half, then only half the amount of potential unbalance remains. If a coupling is balanced without changing tolerances, the unbalance is improved by only 5–20%.

Remember, the tighter the tolerances and the fits, the more expensive the couplings will be. A real-life practical limit is reached at some point. As tolerances are tightened, the price of the coupling not only increases, but interchangeability is lost, delivery time is extended, and in some instances the choice of coupling and potential vendor is limited. An attempt to show how tolerances affect the cost is shown in Figure 3.27.

b. Component Balancing. Component balancing is usually the best way to balance couplings that have inherent clearances between mating parts or require clearances in balancing fixtures. Component balance offers interchangeability of parts, usually without affecting the level of potential unbalance. In most cases, component-balanced couplings can approach the potential unbalance limits of assembly-balanced levels.

	General tolerances[a]	Important tolerances[b]	DBC true location	Approximate cost
Low speed	±1/64	±0.005	±1/64	1
Intermediate speed	±0.005	±0.002	±0.005	1.5–2
High speed	±0.002	±0.001	±0.005	3–4

[a] General tolerance applied to noncritical diameters and lengths (e.g., coupling O.D.).

[b] Important tolerances apply to critical diameters and lengths (e.g., bores, pilots).

FIGURE 3.27 Coupling tolerances versus cost.

This is particularly true when a large, heavy mandrel or fixture must be used to assembly balance a coupling.

c. Assembly Balancing. Assembly balancing of couplings may provide the best overall coupling balance. This is true when no clearance exists between parts (i.e., disc or diaphragm couplings). The balancing fixture and mandrels are lightweight and are either manufactured to extremely tight tolerances [0.0003 in. total indicator readings (TIRs) to 0.0005 in. TIRs; this is 300–500 μin.], or somehow the coupling is locked or rigidized without the use of a fixture or a mandrel. For relatively large assemblies, the mandrels and fixtures may introduce more error than if the coupling was not balanced at all. It is possible to unbalance the coupling more than it was originally unbalanced. Assembly-balanced couplings are matchmarked, and components should not be interchanged or replaced without rebalancing.

d. Field Balancing. On very high speed equipment and/or lightweight equipment, it may not be possible to provide a balanced coupling to meet the requirements due to the inherent errors introduced when a coupling is balanced on a balancing machine. When this occurs, the coupling manufacturer can provide a means for easily adding weight to the coupling. Then the coupling can be balanced by trial and error on the equipment itself. Couplings can be provided with balancing rings that have radial setscrews or tapped holes so that setscrews or bolts with washers can be added or subtracted. There are also other means of facilitating field balancing.

5. Types of Coupling Balance

Most coupling manufacturers can and will supply couplings that are brought into balance. Some coupling manufacturers have what they call standard balancing procedures and practice. Extra charges may be made for balancing procedures other than their standard practices. For example, some coupling manufacturers prefer to component balance only. The different types of coupling balance used by manufacturers are described below.

Today most couplings are supplied *as manufactured* with no balancing.

Controlled tolerances and fits usually provide the most significant improvement in the potential unbalance of couplings. This approach to improving balance also can substantially increase the price of a coupling. It also is the most difficult to explain, so the buyer usually asks that the coupling be balanced anyhow.

Component balance can usually produce potential unbalanced values equal to those of an assembly-balanced coupling. It offers the advantage of making it possible to replace components (as related to balance, some couplings are not interchangeable for other reasons) without replacing the assembly. This technique is shown in Figures 3.28 and 3.29.

Component balance with some subassembly balance can sometimes produce a better balance than an assembly-balanced coupling. This is particularly true of couplings that must have running clearances. One example of this would be gear couplings. If gear couplings are assembly balanced, they are balanced with the radial clearance between the sleeve and hubs tight, and then after balance the tips (usually of the

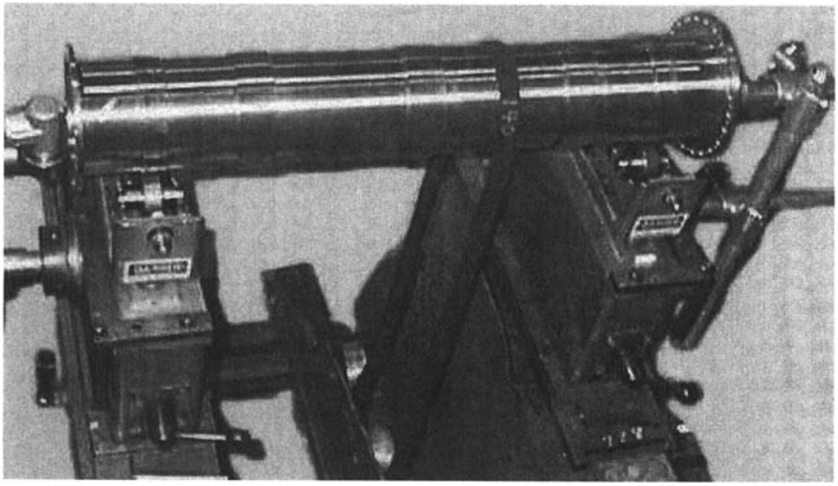


Figure 3.28 Component balancing of geared spacer. (Courtesy of Ameridrives International.)

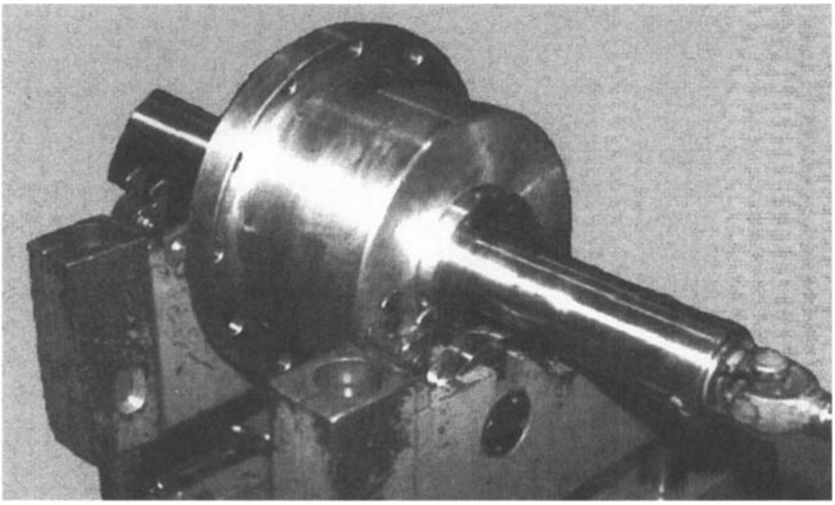


Figure 3.29 Component balancing a rigid coupling (with a mandrel). (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

hub) are machined to provide the proper operational clearance, which results in a balance that is truly upset. If you bolt the sleeves to the spacer, machine rolling bands on the outside diameter of the sleeves (concentric to the gear), and then balance this "subassembly" and the two hubs, you get a better balance than from an assembly-balanced coupling.

Component balance with selective assembly sometimes offers the best possible balance attainable without field balancing on the equipment. Parts are component balanced, and then runouts (TIRs) are checked. The highs of the TIR readings between controlling diameters for mating parts are marked. At final coupling assembly the high spots are assembled 180° out of phase. This tends to negate eccentricities and reduces the potential unbalance of the coupling. The parts are still interchangeable as long as replacement parts are inspected and marked for their TIRs.

Assembly balancing generally offers the best balance, but it usually expensive because it may require mandrels and/or fixtures that are

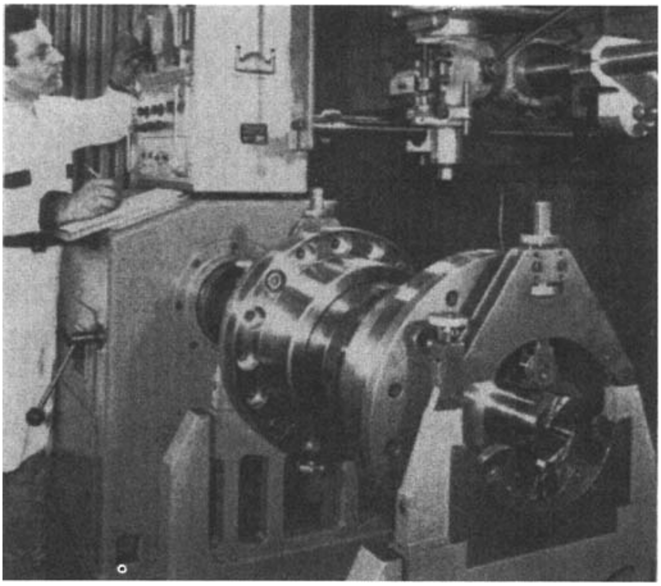


FIGURE 3.30 Assemby balancing of a gear coupling. (Courtesy of Lovejoy, Inc.)

made with extreme accuracy. The coupling is basically rigidized with a mandrel or fixtures and balanced (Figures 3.30 and 3.31). Disc, diaphragm, and some special types of gear couplings can be locked rigid with various locking devices that are usually incorporated into the coupling design (Figure 3.32). The coupling is rolled on rolling surfaces that are aligned or machined to the coupling bores or alignment pilots. This type of balance can usually provide a better balance than can be achieved using a mandrel. This is because there is no added weight to the assembly when it is balanced. On a very large or very long coupling, a mandrel assembly can account for almost half the weight of the coupling. This can introduce significant balancing errors. On assembly-balanced couplings, parts cannot be replaced without rebalancing the coupling.

Field or trim balancing on the equipment offers the best and, of course, usually the most costly way to balance a coupling because of the trial and error and time involved. Usually one is balancing the entire

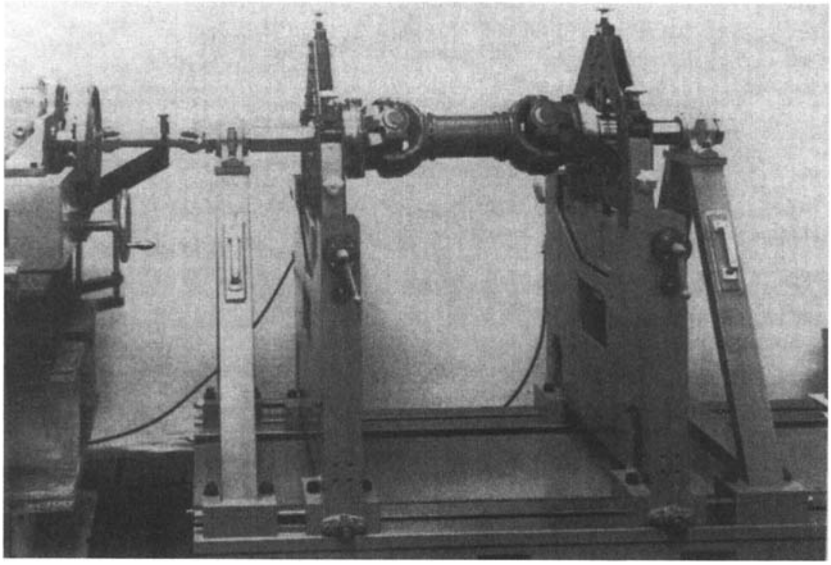


FIGURE 3.31 Assembly balancing of a universal joint. (Courtesy of Spicer Universal Joint Division of Dana Corporation.)

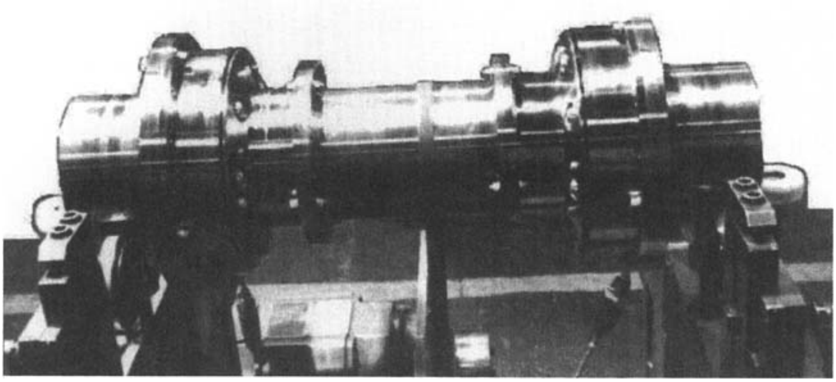


FIGURE 3.32 Assembly balancing of a high-performance disc coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

train by making corrections on the coupling. This makes it impossible to change any part of the coupling or any part of the drivetrain without rebalancing.

D. COUPLING BALANCE SPECIFICATIONS

1. AGMA Specification (AGMA 9000—Flexible Coupling, Potential Unbalance Classification)

The balance limit placed on a coupling should be its potential unbalance limit and not its residual unbalance limit. The residual unbalance limit usually has little to do with the true coupling unbalance (potential). It can be seen in many cases that by cutting the residual unbalance limit in half, the coupling potential unbalance may be changed by only 5%. The best method of determining the potential unbalance of couplings is by calculating the square root of the sum of the squares of the maximum possible unbalances of all the contributing values. These unbalances are mostly from the eccentricities between the coupling parts but include any other factors that produce unbalance.

The coupling balance limit is defined by the American Gear Manufacturers Association (AGMA) as a range of unbalance expressed in microinches (μin.). The potential unbalance limit classes for couplings are listed in Figure 3.33. They are given in maximum root mean square (rms) microinches of displacement of the inertia axis of rotation at the balance plane. Limits are given as displacement per plane.

The residual unbalance limit for a part of an assembly-balanced coupling is shown in Figure 3.34. Tolerances tighter than these usually do very little to improve the overall potential unbalance of a coupling assembly.

2. ISO 1940/1 and ANSI S2.19-1989

These two specifications provide classifications of balance quality that are called grades. They give recommended grades of balance for various types of machinery. Proposed standard ISO 14691 standard ISO 1940/1 for general-purpose applications proposes that couplings for these applications meet ISO 1940 grade 16 or 30 μm, whichever is greater. Also, proposed standard ISO CD 10441 for special-purpose

AGMA coupling balance class	Maximum displacement of principal inertia axis at balancing planes (rms μin.)
4	Over 32,000
5	32,000
6	16,000
7	8,000
8	4,000
9	2,000
10	1,000
11	500

Figure 3.33 AGMA coupling balance classes.

Speed class	Residual unbalance limits (μin.)
Low-speed couplings	500[a]
Intermediate-speed couplings	200
High-speed couplings	50

[a] Low-speed couplings are usually not balanced.

Figure 3.34 Practical residual unbalance limits.

applications proposes that couplings for these applications meet ISO 140 grade 6.3 or 12 μm, whichever is greater. Therefore a pump running at 3600 rpm would be balanced to ISO 16 or 30 μm, whichever is greater, and a compressor running at 6000 rpm would be balanced to meet ISO 140 grade 6.3 or 12 μm, whichever is greater. Figure 3.35 shows speed versus permissible residual mass center displacement.

3. The Arbitrary Balance Criteria

What is meant by the arbitrary balancing criteria? The limits (potential and residual) described in the previous section give the most realistic values for unbalance limits. Many criteria are presently being used in

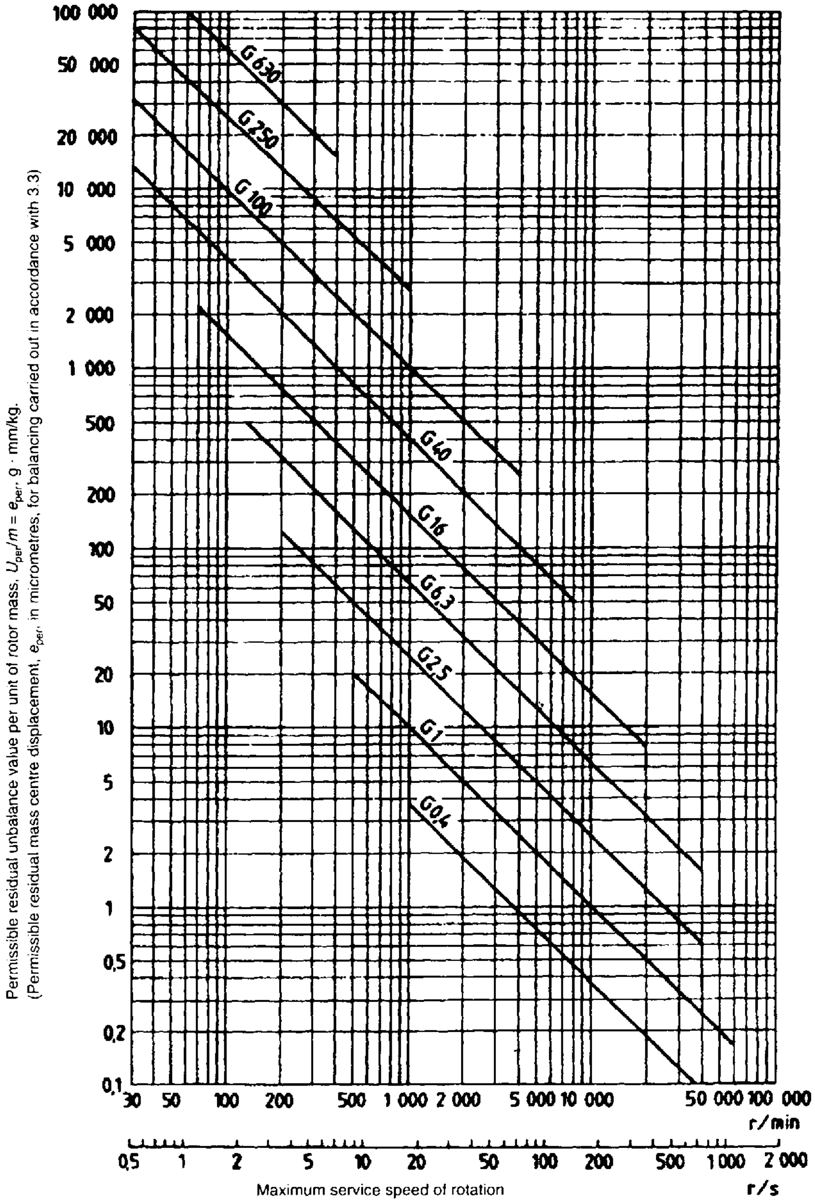


FIGURE 3.35 ANSI S2. 19-1989 (inches). See ISO 1940/1 for micrometers.

several coupling specifications and they are generally referred to as the arbitrary limits. The most common practice is to express the unbalance limit as

$$\text{oz-in. (unbalance)} = \frac{KW}{N} = U \text{ per balance plane} \tag{3.92}$$

where

W = weight of the part per balance plane (lb)
N = operating speed of the coupling (rpm)
K = 40–120 for potential unbalance limits
K = 4–12 for residual unbalance limits

4. API 671 Limits

API defines balance in two classes:

1. Low-speed coupling: speeds less than 1800 rpm, AGMA class 9.
2. High-speed couplings over 1800 rpm (see Figure 3.36).

5. Comparison of the Various Specifications

In general, the limits determined by the various specifications are not too bad, but in some cases a very expensive balanced coupling may result when it is not really necessary. In other cases, specifying tight tol-

Residual limit	Potential limit
U = 4 W/N oz-in.	U = 40 W/N oz-in.
or	or
U = 0.0008W oz-in.	U = 0.008W oz-in.
or	or
U = 0.01 oz-in.	U = 0.1 oz-in.
whichever is greatest	whichever is greatest

FIGURE 3.36 API 671 Limits.

erances will result in arguments, delays, and so on, while everyone involved regroups and tries to place blame for not specifying the limits correctly and/or not balancing the coupling correctly.

If we further expanded the arbitrary potential limits into three coupling speed classes (see Figure 3.37), we could come up with those shown in Figure 3.38.

As stated before, it is best to state unbalance in terms of microinches. If this is done, another arbitrary criterion results, but now if a low limit tolerance is applied (in microinches), the balance limit has some relationship to the tolerances and fits that are used to define the various classes of couplings. Values that result from doing this are shown in Figure 3.39.

Now if we used the same approach and classified residual unbalance limits for each speed class, the limit would be as illustrated in Figure 3.40.

Speed class	Maximum velocity of O.D. of coupling (in./sec)
Low speed	<2300 in./sec
Intermediate speed	>2300 but <3200 in./sec
High speed	>3200 in./sec

Note: where *V* – *O.D.* × RPM/19 in./sec
where O.D. = maximum outside diameter of the coupling (in.).

FIGURE 3.37 Speed classes for couplings

Speed class	Potential unbalance limit (oz-in.)
Low speed	120 W/N oz-in.
Intermediate speed	80 W/N oz-in.
High speed	40 W/N oz-in.

FIGURE 3.38 Aribitrary potential unbalance limits.

Speed class	Potential unbalance limit	AGMA classes (μin.)
Low speed	7,500,000/N or 4000	8
Intermediate speed	5,000,000/N or 2000	9
High speed	2,500,000/N or 500	11

FIGURE 3.39 Potential unbalance limits.

Speed class	Residual unbalance (oz-in.)	Residual unbalance limit (Min.)
Low speed	12 W/N	750,000/N or 400
Intermediate speed	8 W/N	500,000/N or 200
High speed	4 W/N	250,000/N or 50

FIGURE 3.40 Residual unbalance limits.

E. Determining When to Balance and at What Level

The amount of coupling unbalance that can be satisfactorily tolerated by any system is dictated by the characteristics of the specific connected machines and can be determined by detailed analysis or experience. Certain factors must be considered in determining the system's sensitivity to coupling unbalance.

Shaft end deflection. Machines having long and/or flexible shaft extensions are relatively sensitive to coupling unbalance.

Bearing loads relative to coupling weight. Machines having lightly loaded bearings or bearing loads determined primarily by the overhung weight of the coupling are relatively sensitive to coupling unbalance. Machines having overhung rotors or weights are often sensitive to coupling unbalance.

Bearing, bearing supports, and foundation rigidity. Machines or systems with flexible foundations or supports are relatively sensitive to the coupling unbalance.

Machine separation. Systems having long distances between machines often exhibit coupling unbalance problems.

Other factors may influence coupling unbalance sensitivity. The data in Figure 3.41 were taken from the American Gear Manufacturers Association standard AGMA 9000. In general, selection bands can be grouped into the following speed classifications.

Low speed: A and B
Intermediate speed: C, D, and E
High speed: F and G

The speed classifications from AGMA 9000 are also presented in Figure 3.42. The graph has also been extended to 2000 lb.

API 610 couplings and API 671 low-speed couplings (<1800 rpm) fall into selection band D, AGMA 9 for an average system.

API 671 high-speed couplings fall into selection band F, AGMA 11 for an average system.

Proposed ISO 14691 couplings generally fall into selection band C, AGMA 8 for an average system.

Proposed ISO 10441 coupling generally fall into selection band F, AGMA 11 for an average system.

	Typical AGMA coupling balance class: system sensitivity to coupling unbalance		
Selection band	Low	Average	High
A	5	6	7
B	6	7	8
C	7	8	9
D	8	9	10
E	9	10	11
F	10	11	
G	11		

FIGURE 3.41 Coupling balance classes versus selection bands.

F. Summary

Coupling balance requirements and limits are hopefully better understood now than they were a decade or so ago. The balance limits (potential and residual) for couplings should be specified in microinches of mass displacement. This can put balance limits into perspective in terms of manufacturing and requirements based on speed. Specifying limits without analyzing the system's real needs is guesswork and may unnecessarily increase the coupling cost or preclude the use of some perfectly acceptable couplings.

Some important things that should be remembered are that a coupling is a collection of parts that must be disassembled for assembly and/or maintenance of the equipment and maintenance of the coupling itself. The balance level is most affected by the coupling's ability to be repeatedly assembled and disassembled without changing the various mass eccentricities of the coupling parts.

EXAMPLE 1 COUPLING BALANCE SELECTION

The best way to gain an understanding of how to determine the coupling balance requirement is to go through an example case and some calculations.

Conditions. You are a pump manufacturer and have two units that consist of a motor, gear, and pump. You are to supply the couplings between the gears and the pumps.

Operating data. Unit 1 operates at 6000 rpm. Unit 2 operates at 9000 rpm.

Coupling data. You have done your homework and have selected a gear coupling. You have received quotes from the manufacture of brand Y and brand Z. Brand Y and Z couplings both weigh 100 lb. The brand Z manufacturer's quote said that the coupling conforms to an AGMA class 9 balance level. The brand Y manufacturers did not give any information on the balance quality of their coupling so you go back and ask. They then tell you that brand Y also conforms to AGMA class 9.

What level of balance do you need for these couplings? Using Fig-

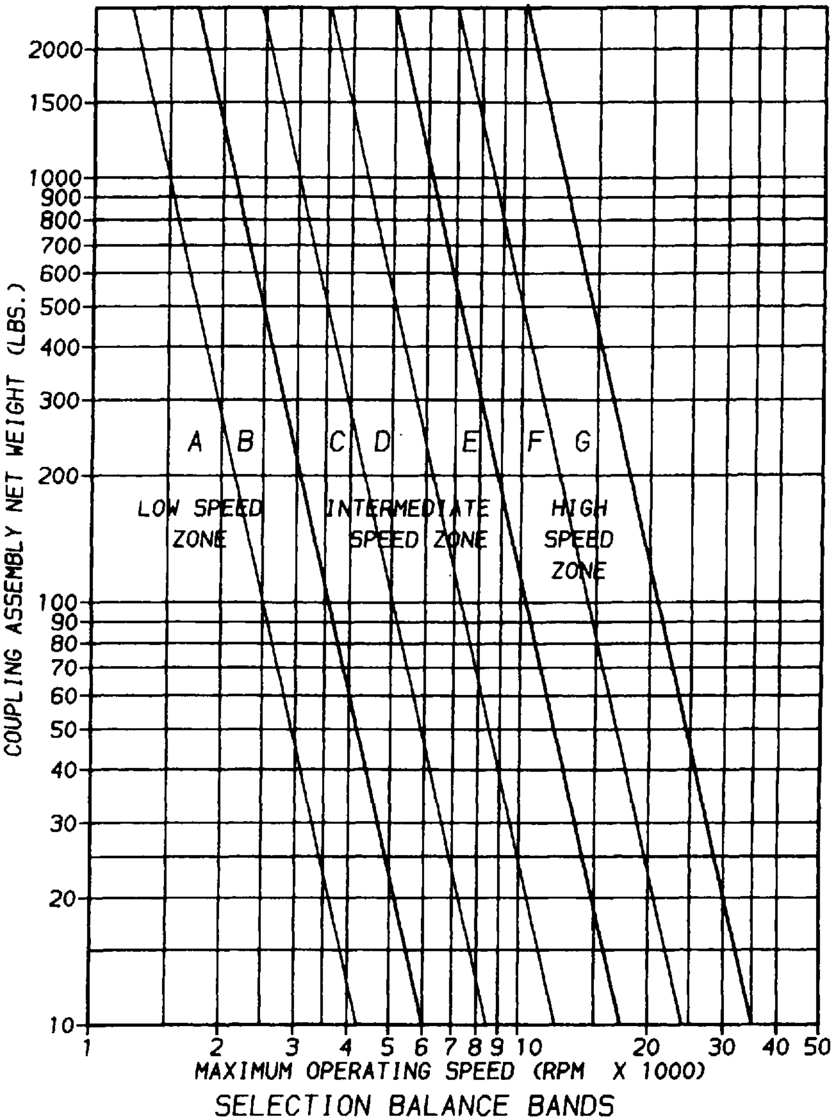


FIGURE 3.42 Balance selection weight versus speed.

ure 3.42, you can find which band these applications fall into. Unit 1 falls into band D. Unit 2 falls into band E.

Next, you go to Figure 3.41 to find which balance class is required. This requires that you decide how sensitive the equipment is. You decide, based on your past experience with similar equipment, that it is a relatively average system. Therefore, you can now determine the coupling balance class required. Unit 1 requires a class 9 or, from Figure 3.43, 2000 μin. Unit 2 requires a balance class 10 or, from Figure 3.43, 1000 μin.

What type of coupling do you need? Unit 1 can use the coupling as originally quoted. Unit 2 needs a class 10.

So you decide that you had better buy a balanced coupling. You go back to the manufacturers of brands Y and Z and ask them to quote on a balanced coupling and tell you what balance class it would be. The new quotes state that these couplings would cost 25–50% more but that the balance class would still be a class 9.

You now need calculations. The brand Y calculations are shown in Examples 2 and 3. The brand Z calculations are shown in Examples 2 and 4. You review the calculations and find that

	Potential unbalance	Improvement (%)
Unbalanced standard tolerances	1765	
Assembly balanced	1456	17.5
Component balanced	1405	20.4
Assembly-balanced tighter tolerances on balance	1452	17.7
Component-balanced tighter tolerances on balance	1402	20.5
Unbalanced tighter tolerances	962	45.5
Assembly-balanced tighter tolerances	943	46.6
Component-balanced tighter tolerance	859	51.3

FIGURE 3.43 Summary of balance selection example.

1. As manufactured, both brand Y and brand Z couplings have a potential unbalance of approximately 1765 μin.
2. An assembly-balanced coupling has a potential unbalance of 1456 μin. (brand Y).
3. A component-balanced coupling has a potential unbalance of 1405 μin. (brand Z).
4. Therefore, it becomes evident that just balancing these couplings will not meet class 10 requirements. In fact, it improves the potential unbalance by only 17–20%.

Now what? Let's get the coupling manufacturers to balance to tighter tolerances. You do this on the pump rotors. Let's plug this into the calculations. Tighter balancing tolerances for an assembly-balanced coupling give 1452; tighter balancing tolerances for a component-balanced coupling give 1402. This cannot be. You have improved balance by only 0.15–0.2%.

This is becoming tedious. So you sit and scratch your head and review all those calculations again. You see that the biggest influence on the coupling potential unbalance are the eccentricities and the pilot clearances inherent in these couplings.

Let's see what happens if you cut these tolerances in half. Coupling not balanced, 962 μin.; coupling assembly balanced, 943 μin.; coupling component balanced, 859 μin.

Get quotes for tighter-toleranced couplings. You ask for quotes on couplings that meet AGMA class 10 (these would be couplings built with tighter tolerances). The quote comes back with couplings costing 200–300% more. However, what choice do you have?

You review the calculations (see Example Table 1). From these it is clear that you really do not need to balance this new type of coupling, but you are going to have trouble explaining this. So you buy a component-balanced coupling made to tighter tolerances and hope nobody asks you why a balanced coupling cost so much.

This example is meant to give you a better understanding of coupling balance requirements and limits. The balance limits (potential or residual) for a coupling should be specified in microinches of mass displacement. This helps put balance limits into a real-world perspective.

Example Table 1 Summary of coupling balance selection example

	Potential unbalance	Improvement
Unbalanced standard tolerances	1765	
Assembly-balanced	1456	17.5%
Component-balanced	1405	20.4%
Assembly-balanced tighter tolerances on balance	1452	17.7%
Component-balanced tighter tolerances on balance	1402	20.5%
Unbalanced tighter tolerances	962	45.5%
Assembly-balanced tighter tolerances	943	46.6%
Component-balanced tighter tolerance	859	51.3%

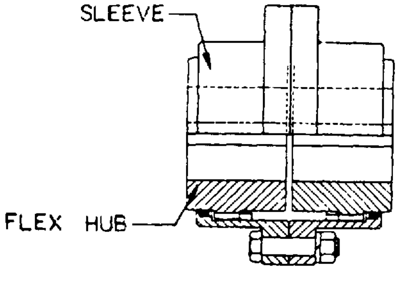
Specifying arbitrary limits without analyzing the system's real needs may increase the coupling cost unnecessarily or preclude the use of some perfectly acceptable couplings.

Again, remember that a coupling is an assembly of parts that must be taken apart for assembly, maintenance of equipment, and self-maintenance. Its balance level is most affected by its ability to be assembled and disassembled repeatedly without changing the various mass eccentricities of its parts.

EXAMPLE 2 UNCORRECTED GEAR COUPLING

coupling geometry and assumed data (see Example Figure 1):

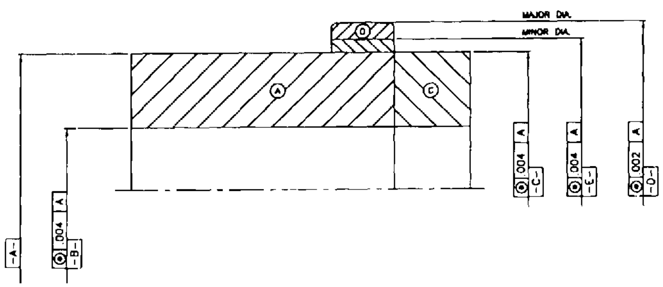
1. Weights
   a. Coupling assembly = 100 lb
   b. Coupling component weights
      Hubs: 2 @ 23 lb = 46 lb
      Sleeves: 2 @ 26 lb = 52 lb
      Hardware: 1 set = 2 lb
2. Coupling pilot surface eccentricity: Maximum eccentricity of hub major diameter = 0.004 in. TIR



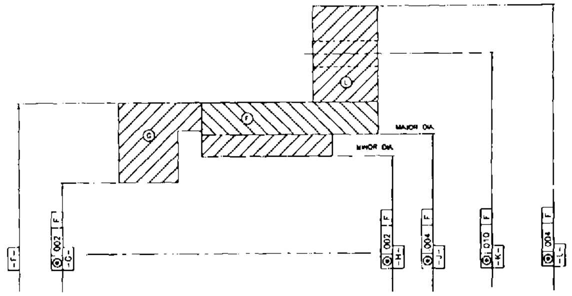
Example Figure 1.

3. Coupling pilot surface clearance: Hub major diameter to sleeve root clearance (radial) = 0.0015 in.
4. Inherent unbalance of coupling components: Potential inherent unbalance of the hub and sleeve components is calculated for the assumed manufacturing tolerances shown on the sketches of the components. The fasteners are assumed to have a uniform weight but are installed in holes that allow them to be eccentric by one-half the eccentricity shown for datum *K*. This includes 0.005 in. inaccuracy in the bolt hole position.

Data for calculated potential unbalance of coupling hub (see Example Figures 2 and 3) are given in Example Tables 2 and 3: and the unbalance per correction plane, in Example Table 4.



Example Figure 2.



Example Figure 3.

Example Table 2 Potential Unbalance of Hub (Example 2)

Datum	Maximum runout to *B* (TIR)	Maximum eccentricity to *B*	Weight of section (lb)	Maximum potential unbalance oz-in.	Maximum potential unbalance (oz-in.)^2
A	0.004	0.002	12	0.384	0.147
B	—	—	—	—	—
C	0.008	0.004	4	0.256	0.065
D	0.004	0.002	4	0.128	0.0164
E	0.004	0.004	3	0.192	0.0368
Totals				0.96	0.265

Rms value of potential unbalance of hub = $\sqrt{0.265}$ = 0.5147 oz-in.

EXAMPLE 3: GEAR COUPLING BALANCED AS AN ASSEMBLY

Coupling geometry and assumed data are given in Example Figure 4 and calculated value in Example Table 5.

1. Weights
   a. Total coupling weight = 100 lb
   b. Coupling component weights
      Flex hubs: 2 @ 23 lb = 46 lb
      Sleeves: 2 @ 26 lb = 52 lb
      Hardware: 1 set = 2 lb

continues on p. 148

EXAMPLE TABLE 3 Potential unbalance of sleeve and hardware (Example 2)

Datum	Maximum runout to *J* (TIR)	Maximum eccentricity to *J*	Weight of section (lb)	Maximum potential unbalance oz-in.	Maximum potential unbalance (oz-in.)^2
F	0.004	0.002	8	0.256	0.065
G	0.006	0.003	5	0.240	0.057
H	0.006	0.003	3	0.144	0.021
J	—	—	—	—	—
K	0.014	0.007	1[a]	0.112	0.0126
L	0.008	0.004	10	0.64	0.4096
Totals				1.402	0.565

Rms value of potential unbalance of sleeve and hardware = $\sqrt{0.565}$ = 0.752 oz-in.

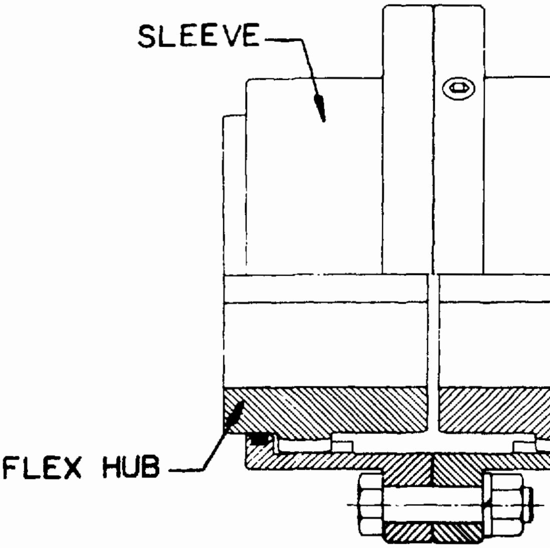
[a] One pound is one-half the weight of the hardware.

EXAMPLE TABLE 4 (Example 2)

Calculation	Unbalance per correction plane oz-in.	Unbalance per correction plane (oz-in.)^2
Inherent unbalance of uncorrected coupling		
Hub	0.5147	0.265
Sleeve and hardware	0.752	0.565
Coupling pilot surface eccentricity	0.864[a]	0.746
Coupling pilot surface clearance	0.648[b]	0.419
Hardware unbalance	—	—
Others	—	—
Totals	2.778	1.995

Rms value of potential unbalance = $\sqrt{1.995}$ = 1.412 oz-in.
Rms displacement of principal inertia axis = 1.412/(50 × 16) = 1765 μin.

[a] Runout of hub major diameter: (26 + 1) (16) (0.004/2)
[b] Tooth pilot clearance (radial) after mounting hub: (26 + 1) (16) (0.0015)



Example Figure 4.

   c. Balancing tooling weights
      Mandrel = 30 lb
      Bushings: 2 @ 2 lb = 4 lb
2. Balance correction tolerance = 100 μin. per plane
3. Balance machine error
   a. Sensitivity = 10 μin. per plane
   b. Driver error assumed to be negligible
4. Balancing mandrel assembly unbalance = 10 μin. per plane.
5. Dial indicator graduation = 0.0001 in.
6. Mandrel assembly mounting surface eccentricity
   a. Mandrel runout at mounting surface for bushing = 0.0005 in. TIR
   b. Bushing bore to bushing mounting surface for hub runout = 0.0004 in. TIR
   c. Mandrel-to-bushing clearance = 0.0005 in.
   d. Bushing mounting surface to hub bore clearance = 0.0005 in.
7. Coupling pilot surface eccentricity: It is assumed that parts

Example Table 5 Calculations for Example 3

	Unbalance per correction plane	
Calculation	oz-in.	(oz-in.)^2
Balance correction tolerance		
$\frac{100}{2} \times 16 \times 0.0001$	0.08	0.0064
Balance machine error		
Sensitivity		
$\frac{134}{2} \times 16 \times 0.00001$	0.01072	0.0001149
Driver error assumed to be negligible		
Balancing mandrel assembly unbalance		
Mandrels	n.a.	n.a.
Bushings	n.a.	n.a.
Mandrel assembly		
$\frac{30 + 2 + 2}{2} \times 16 \times 0.00001$	0.00272	0.0000074
Mandrel assembly mounting surface eccentricity		
Mandrel runout		
$\frac{100}{2} + 2 \times 16 \times \frac{0.0001 + 0.0005}{2}$	0.2496	0.0623
Mandrel-to-bushing clearance		
$52 \times 16 \times \frac{0.0005}{2}$	0.208	0.043
Bushing bore-to-mounting surface runout		
$\frac{100}{2} \times 16 \times \frac{0.0001 + 0.0004}{2}$	0.200	0.0400
Bushing-to-hub bore clearance		
$\frac{100}{2} \times 16 \times \frac{0.0005}{2}$	0.200	0.040
Coupling pilot surface eccentricity (see assumed data, item 7)	—	—
Coupling pilot surface clearance		
Tooth tip clearance		
$\frac{26 + 26 + 2}{2} \times 16 \times \frac{0.003}{2}$	0.648	0.419

EXAMPLE TABLE 5 Continued

	Unbalance per correction plane	
Calculation	oz-in.	(oz-in.)^2
Eccentricity of hub after remachining		
$(26 + 1) \times 16 \times \frac{0.004}{2}$	0.862	0.746
Hardware unbalance		
$\frac{0.2\text{ g } 4.875}{28.35} \times \frac{\sqrt{2}}{2}$	0.0244	0.00059
Others	0	0
Totals	2.485	1.357

Rms value of potential unbalance = $\sqrt{1.357}$ = 1.165 oz-in.
Rms displacement of principal inertial axis = 1.165 / (50 × 16) = 1456 μin.

are match-marked and therefore the balancing operation compensates for these runouts (hub bore to tooth tip and sleeve root to sleeve pilot for each sleeve). After balancing, the tips of hubs are machined to hub bore to 0.004 TIR.

8. Coupling pilots surface clearance:
   a. Tooth tip pilot clearance = 0.003 after balancing, 0.000 before balancing
9. Hardware unbalance:
   a. Fasteners (deviation per set) = 0.2 g
   b. Bolt circle diameter = 9.75 in.

EXAMPLE 4. GEAR COUPLING BALANCED AS INDIVIDUAL COMPONENTS

Coupling geometry and assumed data are given in Example Figure 4 and calculated value in Example Table 6.

1. Weights
   a. Total coupling weight = 100 lb
   b. Coupling component weights
      Flex hubs: 2 @ 23 lb = 46 lb

continues on p. 152

EXAMPLE TABLE 6 Calculations for Example 4

Calculation	Unbalance per correction plane	
	oz-in.	(oz-in.)^2
Balance correction tolerance		
$\frac{100}{2} \times 16 \times 0.0001$	0.08	0.0064
Balance machine error		
Sensitivity		
$\frac{100}{2} \times 16 \times 0.00001$	0.008	0.00006
Indicating surface eccentricity		
Hub bore		
$23 \times 16 \times \frac{0.0001 + 0.001}{2}$	0.2024	0.0409
Sleeve major diameter		
$26 \times 16 \times \frac{0.0001 + 0.001}{2}$	0.2288	0.0523
Coupling pilot surface eccentricity		
Runout of hub major diameter		
$(26 + 1) \times 16 \times \frac{0.004}{2}$	0.864	0.746
Coupling pilot surface clearance		
Tooth pilot clearance		
$(26 + 1) \times 16 \times \frac{0.003}{2}$	0.648	0.419
Hardware unbalance		
$\frac{0.2\text{ g} \times 4.875}{28.35} \times \frac{\sqrt{2}}{2}$	0.0244	0.00059
Others		
Totals	2.055	1.265

Rms value of potential unbalance = $\sqrt{1.265}$ = 1.124 oz-in.
Rms displacement of principal inertia axis = 1.124 / (50 × 16) = 1405 μin.

Sleeves: 2 @ 26 lb = 52 lb
Hardware: 1 set = 2 lb

2. Balance correction tolerance = 100 μin. per plane
3. Balance machine error
   a. Sensitivity = 10 μin. per plane
   b. Driver error assumed to be negligible
4. Dial indicator graduation = 0.0001 in.
5. Coupling pilot surface eccentricity: Maximum eccentricity of hub major diameter = 0.004 in. TIR
6. Coupling pilot surface clearance: Tooth tip pilot clearance = 0.003 in.
7. Hardware unbalance
   Fasteners (deviation per set) = 0.2 g
   Bolt circle diameter = 9.75 in.

IV. INTERFACE CONNECTIONS

One area that is often overlooked or ignored because everyone feels that it is someone else's responsibility is the interface connection. What is an interface? It is the torque–transmission connection between the equipment and the coupling. As stated in Section 3.1, the responsibility for the interface connection must lie with the coupling selector (equipment manufacturer or user).

There are two basic types of interface connections, the cylindrical bore and the flange connection. Figure 3.44 lists some of the most common interfaces used. Shown are the relative torque capacity, ease of assembly/disassembly, reliability, machining requirements, usage, and relative cost of each. The most common type of interface connection is the cylindrical bore with one or more keys and an interference fit between the shaft and the coupling hub. The trend today in high-speed equipment indicates a preference for tapered hydraulic hubs. This section covers the interface connections listed in Figure 3.44, together with their usage, application, and standards for each (where they exist).

A. Shaft Connections

1. Clearance-fitted Bores

Clearance-fitted bores are a type of interface connection that is very commonly used with small-horsepower couplings, usually under 150 hp. Some very small couplings (miniatures) may use just a setscrew to hold the hub on the shaft (both torsionally and axially). On larger units, however, the clearance fit is used in conjunction with a key and two setscrews.

The most common standard available for clearance fits for couplings is AGMA 9002. This specification has two classes of clearance-fitted bores (see Figure 3.45):

Class 1: This type of fit is more expensive and requires tighter control in shaft and coupling bore manufacture.

Class 2: This type of fit is usually cheaper and provides almost twice the clearance of class 1.

2. Interference-fitted Bores

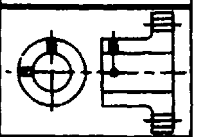
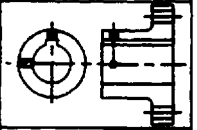
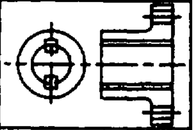
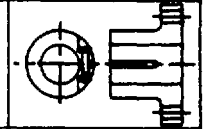
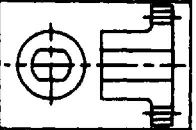
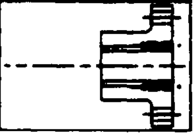
Interference-fitted bores are the most commonly used interface connections, particularly for medium-sized couplings, and are generally used for 150–100,000 hp equipment. They are typically used with one or two keys. If the interference is high enough, no key may be required (see Section IV.A.9). Interference-fitted bores minimize fretting corrosion of the hub bore surface that sometimes occurs with clearance-fitted bores.

The most common standard available for interference fits (with keys) for couplings is AGMA 9002, from which Figure 3.45 is extracted. For larger bore sizes than in AGMA, a nominal interference of 0.0005 in. per inch of shaft diameter is commonly used for medium-carbon steel hubs with keys. For the alloy hubs (4140, 4340, etc.) usually used on high-speed couplings, the interference typically ranges from 0.00075 to 0.001 in./in. shaft diameter.

3. Setscrews

Setscrews are usually used to retain clearance-fitted bores (see Figure 3.44A and B), which common have two setscrews, one over the key (if used) and the other 90° or 120° from it. Sometimes the shaft is flattened in the area of the setscrew to help increase the holding power. Setscrews can dig into a shaft and cause assembly and disassembly problems. To help minimize gouging of the shaft, brass setscrews or setscrews with brass tips are sometimes used.

The setscrew interface connection is relatively easy to install and remove as long as care is exercised to prevent galling of the shaft from

	TYPE	TORQUE CAPACITY	EASE OF ASSY. & DISASSY.	RELIABILITY	MACHINING REQUIREMENTS	USAGE	RELATIVE COST	DRAWING
A	CYLINDRICAL BORES WITH CLEARANCE & SETSCREWS.	POOR	EXCELLENT	?	LITTLE	COMMON (SMALL CPLGS.)	LOW	
B	CYLINDRICAL BORES WITH CLEARANCE, KEYWAY, & SET-SCREWS.	GOOD	GOOD	GOOD	SOME	VERY COMMON (SMALL CPLGS.)	MODERATE	
C	CYLINDRICAL BORES WITH INTERFERENCE & KEYWAYS.	VERY GOOD	FAIR	VERY GOOD	MODERATE	MOST COMMON (ALL CPLGS.)	MODERATE	
D	SPLIT HUBS	FAIR	EXCELLENT	?	HIGH	LITTLE to COMMON. (VERY SMALL CPLGS.)	MODERATE to HIGH	
E	FLATTED BORES	VERY GOOD	VERY GOOD	GOOD	MODERATE to HIGH	LITTLE (LARGE CPLGS.)	HIGH	
F	BUSHINGS	FAIR	VERY GOOD	GOOD	MODERATE	COMMON (LOW HP)	MODERATE to HIGH	

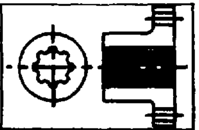
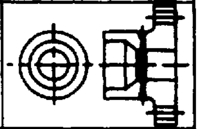
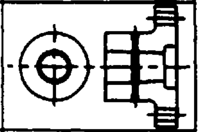
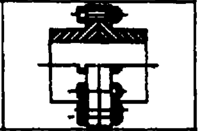
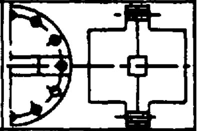
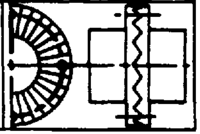
G	SPLINES	EXCELLENT	GOOD	VERY GOOD	HIGH	LITTLE to COMMON	HIGH	
H	KEYLESS STRAIGHT BORES.	VERY GOOD	FAIR	VERY GOOD	MODERATE	LITTLE	MODERATE	
I	KEYLESS TAPERED BORES.	VERY GOOD	FAIR to GOOD	VERY GOOD	HIGH	BECOMING COMMON	HIGH	
J	BOLTED FLANGE	GOOD	GOOD	GOOD	MODERATE	VERY COMMON	MODERATE	
K	FACE KEYS	VERY GOOD	VERY GOOD	VERY GOOD	MODERATE	LITTLE	MODERATE to HIGH	
L	FACE SPLINES	EXCELLENT	VERY GOOD	EXCELLENT	HIGH	LITTLE	HIGH	

FIGURE 3.44 Comparision of coupling attachment methods.

Nominal Bore Range(1)		Clearance Fits					Interference Fits				Nominal Keyway Dimensions(3)			
			Class 1		Class 2						Square		Rectangular	
Over	To (Incl.)	Shaft Tol.(2) (Minus)	Bore Tol. (Plus)	Fit Tol. (Plus)	Bore Tol. (Plus)	Fit Tol. (Plus)	Bore Tolerance Range (Minus)		Fit Tolerance Range (Minus)		Width	Depth	Width	Depth
0.4375	0.5625	0.0005	0.0010	0.0015	0.0020	0.0025	0.0005	0.0010	0.0000	0.0010	0.1250	0.0625	0.1250	0.0468
0.5625	0.8750	0.0005	0.0010	0.0015	0.0020	0.0025	0.0005	0.0010	0.0000	0.0010	0.1875	0.0937	0.1875	0.0625
0.8750	1.2500	0.0005	0.0010	0.0015	0.0020	0.0025	0.0005	0.0010	0.0000	0.0010	0.2500	0.1250	0.2500	0.0937
1.2500	1.3750	0.0005	0.0010	0.0015	0.0020	0.0025	0.0005	0.0010	0.0000	0.0010	0.3125	0.1562	0.3125	0.1250
1.3750	1.5000	0.0005	0.0010	0.0015	0.0020	0.0025	0.0005	0.0010	0.0000	0.0010	0.3750	0.1875	0.3750	0.1250
1.5000	1.7500	0.0010	0.0010	0.0020	0.0020	0.0030	0.0010	0.0020	0.0000	0.0020	0.3750	0.1875	0.3750	0.1250
1.7500	2.0000	0.0010	0.0010	0.0020	0.0020	0.0030	0.0010	0.0020	0.0000	0.0020	0.5000	0.2500	0.5000	0.1875
2.0000	2.2500	0.0010	0.0015	0.0025	0.0020	0.0030	0.0010	0.0020	0.0000	0.0020	0.5000	0.2500	0.5000	0.1875
2.2500	2.7500	0.0010	0.0015	0.0025	0.0020	0.0030	0.0010	0.0020	0.0000	0.0020	0.6250	0.3125	0.6250	0.2187
2.7500	3.0000	0.0010	0.0015	0.0025	0.0020	0.0030	0.0010	0.0020	0.0000	0.0020	0.7500	0.3750	0.7500	0.2500
3.0000	3.2500	0.0010	0.0015	0.0025	0.0030	0.0040	0.0015	0.0030	0.0005	0.0030	0.7500	0.3750	0.7500	0.2500
3.2500	3.7500	0.0010	0.0015	0.0025	0.0030	0.0040	0.0015	0.0030	0.0005	0.0030	0.8750	0.4375	0.8750	0.3125
3.7500	4.0000	0.0010	0.0015	0.0025	0.0030	0.0040	0.0015	0.0030	0.0005	0.0030	1.0000	0.5000	1.0000	0.3750
4.0000	4.5000	0.0010	0.0015	0.0025	0.0040	0.0050	0.0020	0.0035	0.0010	0.0035	1.0000	0.5000	1.0000	0.3750
4.5000	5.0000	0.0010	0.0015	0.0025	0.0040	0.0050	0.0020	0.0035	0.0010	0.0035	1.2500	0.6250	1.2500	0.4375
5.0000	5.5000	0.0010	0.0015	0.0025	0.0040	0.0050	0.0025	0.0040	0.0015	0.0040	1.2500	0.6250	1.2500	0.4375
5.5000	6.5000	0.0010	0.0015	0.0025	0.0040	0.0050	0.0025	0.0040	0.0015	0.0040	1.5000	0.7500	1.5000	0.5000
6.5000	7.0000	0.0010	——	——	——	——	0.0025	0.0040	0.0015	0.0040	1.7500	0.8750	1.7500	0.7500
7.0000	7.5000	0.0010	——	——	——	——	0.0030	0.0050	0.0020	0.0050	1.7500	0.8750	1.7500	0.7500
7.5000	8.0000	0.0010	——	——	——	——	0.0030	0.0050	0.0020	0.0050	2.0000	1.0000	2.0000	0.7500
8.0000	9.0000	0.0010	——	——	——	——	0.0035	0.0055	0.0025	0.0055	2.0000	1.0000	2.0000	0.7500

(1) Preferred Nominal Shaft Sizes:
0.500, 0.625, 0.750, 0.875, 0.9375, 1.000, 1.125, 1.1875, 1.250, 1.375, 1.4375, 1.500, 1.625, 1.750, 1.875, 1.9375, 2.000, 2.125, 2.250, 2.375, 2.4375, 2.500, 2.625, 2.750, 2.875, 2.9375, 3.000, 3.250, 3.375, 3.500, 3.625, 3.750, 4.000, 4.250, 4.500, 4.750, 5.000, 5.250, 5.500, 5.750, 6.000, 6.250, 6.500, 6.750, 7.000

(2) Agrees with NEMR standard MG 1-4.05, June 1981, thru 6.5000.

(3) Non-shaded areas are preferred.

Examples of Tolerancing Bores and Shafts:

2" Nominal bore — Class 1 Clearance Fit			2" Nominal bore — Class 2 Clearance Fit			2" Nominal bore — Interference Fit		
Bore Size	—	2.000 to 2.001	Bore Size	—	2.000 to 2.002	Shaft Size	—	2.000 to 1.999
Shaft Size	—	2.000 to 1.999	Shaft Size	—	2.000 to 1.999	Bore Size	—	1.998 to 1.999
Resulting Fit		Loose 0.000 to 0.002	Resulting Fit		Loose 0.000 to 0.003	Resulting Fit		Tight 0.002 to 0.000

Extracted from AGMA Standard for Bores and Keyways for Flexible Couplings (Inch Series) (AGMA 9002-A86), with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia, 22134.

FIGURE 3.45 Shaft-to-hub fits and nominal keyway sizes (inches).

the setscrews. The setscrews should be secured with Nylok inserts or a locking compound so that they do not work loose. The lock keeps the coupling from spinning on the shaft and ruining it and prevents the setscrew, and possibly the coupling, from becoming a flying projectile.

4. Split Hubs

Split hubs (see Figure 3.44D) are made with a clearance fit between the shaft and the hub bore. The hub is clamped and deformed by one or more bolts and provides an interference fit when installed to transmit torque and axially fix the hub on the equipment shaft. The split hub design makes it relatively easy to assemble and disassemble a coupling. You just loosen the bolt(s) and you have clearance to slide the coupling hub off the shaft.

This type of interface is usually used with relatively small couplings that transmit less than 150 hp. The hubs are usually complex and expensive to make. A hub's torque capacity is usually reduced, making it prone to failure. Due to its configuration, the hub usually becomes the weak link in the chain—in this case, the drivetrain.

5. Keys

Most couplings are supplied with hubs that are bored and keyed (see Figure 3.44C). Coupling manufacturers do not normally supply the keys used at the equipment interface. They are usually supplied and fitted to the equipment and the coupling by the equipment manufacturer. Low-speed and intermediate-speed couplings are usually supplied with one keyway. High-speed and high-torque couplings are usually supplied with two keyways.

The better the fit between the key and the keyways, the more torque this interface connection can handle. Providing for two keyways, which may fit sloppily, does not necessarily increase the torque capacity of the interface connection. Improper fit of keys to keyways can lead to key roll and eventually to splitting of the coupling hub (see Figure 3.46). The combination of tight-fitting keys and an interference-fitted bore provides for an interface connection with high reliability and high torque capabilities.

The most common standards used for key sizes, key fit, keyway dimensions, and tolerances are AGMA's. The following material has been adapted from AGMA standards that define the recommended sizes and tolerances of keys and keyways used with industrial-type flexible couplings. The keyway tolerances and recommended sizes contained in these standards are intended for single-key applications only, but they may be used for multikey connections where application conditions permit.

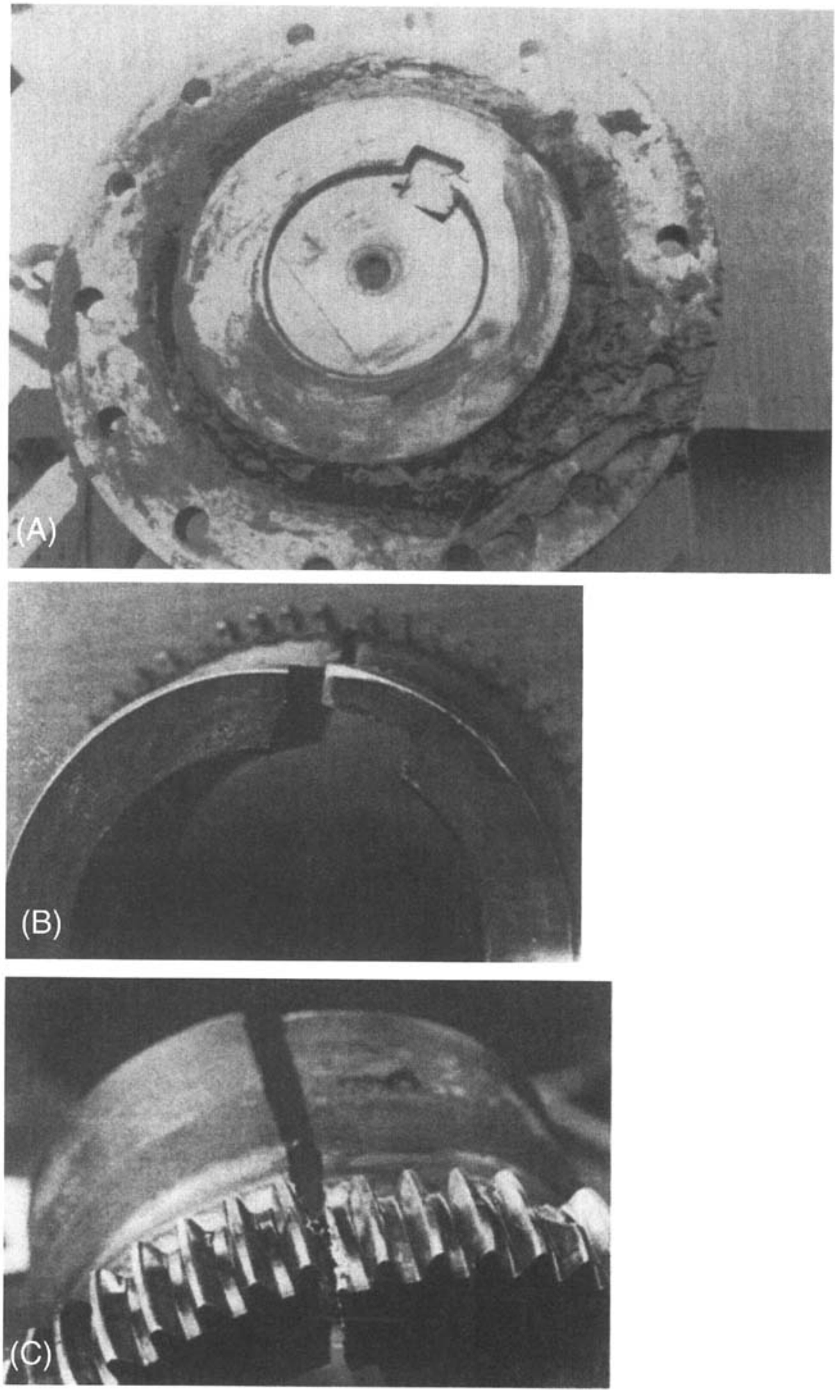


FIGURE 3.46 Hub damage due to improperly fitted keys. (A) Rolled-over key, (B), (C) Split hub.

a. Types of Keys and Keyways. The tolerances in these standards apply to keyways for

Square parallel keys
Rectangular (flat) keys
Plain taper keys

b. Shaft Diameter Range. These standards cover the keyway tolerances and recommended key sizes and tolerances for shafts ranging in diameter from 5/16 in. through 7 in.

c. Class of Key Fit. These standards recognize three classes of key fit:

1. Commercial
2. Precision
3. Fitted

Commercial. Commercial keys, used for most applications, have a clearance fit with the sides of the keyway and use commercial bar stock keys having widths ranging from exact size to minus tolerances. Keyway width and depth tolerances, recommended key tolerances, and the resulting fit dimensions are listed in Figure 3.47.

Precision. Precision keys have a transitional fit with the sides of the keyway and limited clearance over the top of the key. They require the use of keystock having widths ranging from exact size to plus tolerances. For keyway width and depth tolerances, see Figure 3.47.

Fitted. Fitted keys require the use of an oversized key whose width is fitted to suit the keyway at assembly. The height may also be fitted where required by operating conditions. Keyway width and depth tolerances are shown in Figure 3.47.

d. Keyway Dimensions Keyway dimensions for square, rectangular, taper, and Woodruff keys are shown in Figure 3.48 and defined below:

$$C = 0.5D - \sqrt{(0.5D)^2 - (0.5W_2)^2}$$

where

C = chord height at keyway
D = diameter of bore
W_2 = width of keyway

Nominal key width (Wk)			Commercial class fit			Precision class fit		
Over	To (incl.)	Keyway width tolerance	Recommended key tolerance width and height	Resulting side fit[a]	Recommended keyway depth tolerance	Recommended key tolerance width and height	Resulting side fit[a]	Recommended keyway depth tolerance
	5/16	+0.002 −0.000	+0.000 −0.002	CL 0.004 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.002 Tight 0.001	+0.005 −0.000
5/16	1/2	+0.0025 −0.000	+0.000 −0.002	CL 0.0045 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.0025 Tight 0.001	+0.005 −0.000
1/2	3/4	+0.003 −0.000	+0.000 −0.002	CL 0.005 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.003 Tight 0.001	+0.005 −0.000
3/4	1	+0.003 −0.000	+0.000 −0.003	CL 0.006 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.003 Tight 0.001	+0.005 −0.000
1	1 1/2	+0.0035 −0.000	+0.000 −0.003	CL 0.0065 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.0035 Tight 0.001	+0.005 −0.000
1 1/2	2	+0.004 −0.000	+0.000 −0.005	CL 0.009 0.000	+0.016 −0.000	+0.001 −0.000	CL 0.004 Tight 0.001	+0.005 −0.000

Note: Sides of keyways to be parallel within the width tolerance—bottoms of keyways are to be parallel with centerline of bore (or bottom of bore opposite keyway) within the depth tolerance.

[a] CL, Clearance.

Extracted from AGMA Standard for Bores and Keyways for Flexible Couplings (Inch Series) (AGMA 9002-A86), with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia, 22134.

FIGURE 3.47 Keyway width and depth tolerances and width fit.

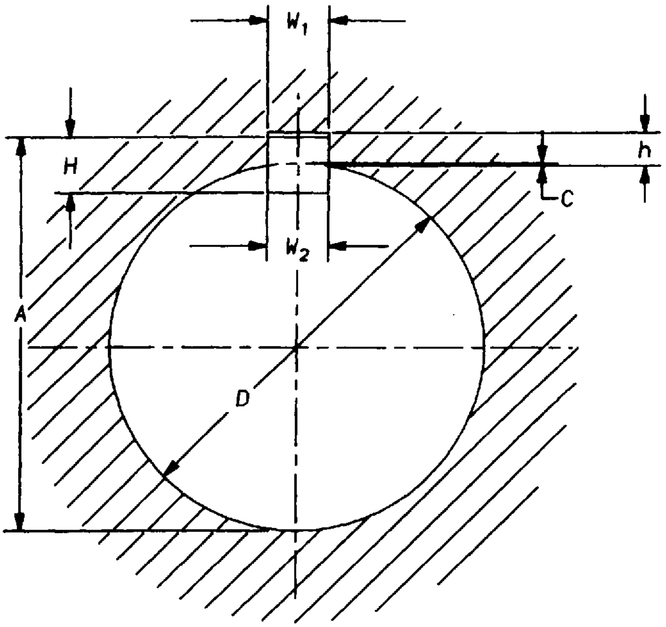


FIGURE 3.48 Key and keyway dimensions.

Keyway width. For square, rectangular, and taper keys and for all classes of fit,

$$W_1 = W_2$$

where W_1 is the width of the key.

Keyway depth

1. For square, rectangular, and Woodruff keys and for all classes of fit,

   $$h = 0.5H$$
   $$A = D + 0.5H - C$$

   where

   A = dimension from top of keyway to opposite side of bore

   H = depth of key

   h = depth of keyway

2. For taper keys and for all classes of fit (measured at large end):

   $$h = 0.5H - 0.020 \text{ in.}$$
   $$A = D + 0.5H - C - 0.020 \text{ in.}$$

e. Keyway Tolerances. Keyway tolerances are illustrated in Figure 3.49.

Keyway offset. Maximum offset (N) for keyways is determined as follows (see Figure 3.49):

$$N = \frac{6 + W_2}{1000} \quad \text{for precision class}$$

$$N = \frac{10 + W_2}{1000} \quad \text{for commercial and fitted classes}$$

Keyway lead (see Figure 3.49B). Permissible lead (J) for keyways must not exceed the values tabulated in Figure 3.50.

Keyway parallelism (see Figure 3.49C). Parallelism is restricted by the keyway width and depth tolerances shown in Figure 3.47.

Finish of keyways. In general, good machine shop practice is indicated for the finish on keyways. It is recommended that keyways and sides and bottoms be finished to a maximum of 125 μin. or better.

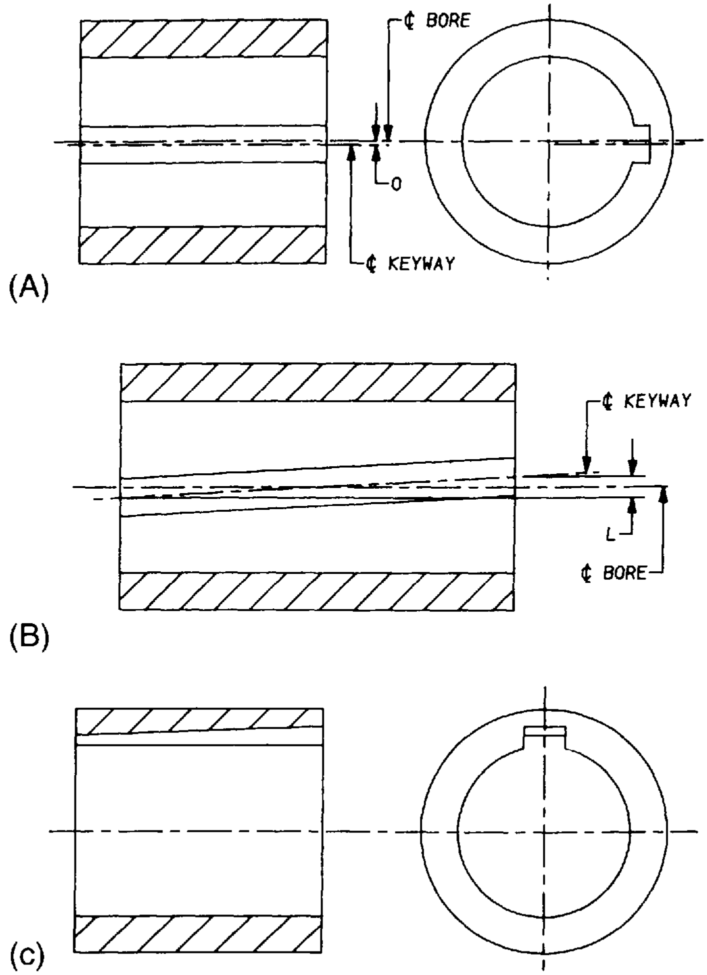


Figure 3.49 Keyway errors. Keyway offset. Key lead. Keyway parallelism.

Keyway length (in.)		Maximum lead	
Over	To (incl.)	Commercial and fitted class	Precision class
—	2	0.002 in.	0.001 in.
2	10	0.001 in./in. of length	0.0005 in./in. of length
10	—	0.010 in.	0.005 in.

Figure 3.50 Permissible keyway lead error values.

Keyway depth (in.)		Keyway fillet radius	Suggested key chamfer (45°)
Over	To (incl.)		
—	1/8	1/64	1/32
1/8	1/4	1/32	3/64
1/4	1/2	1/16	5/64
1/2	7/8	1/8	5/32
7/8	1 1/4	3/16	7/32

Figure 3.51 Values for keyway fillet radius.

Fillet radii in keyways. Couplings may or may not be furnished with fillet keyways. Where fillets are furnished, they will be in accordance with Figure 3.51 unless otherwise specified. Keys must be chamfered or rounded to clear the fillet radii.

6. Flattened Bores

a. Flattened Cylindrical Bores. This type of interface connection (see Figure 3.44E) is usually used for equipment or couplings that are removed or disassembled quite often. It is very common on steel mill drives, where couplings connect to the rolls and the rolls are constantly being changed. The use of the flattened bore for roll necks is shown in Figure 3.44E. Since the bore-to-shaft fit is usually quite loose, it is very common to provide lube fittings in the hub through which lubricants can be introduced, thus helping to minimize wear. Usually, replaceable wear plates are provided with this type of interface connection.

b. Square Bores. This type of interface connection is commonly used on small steel mill drives and on universal joints for agricultural equipment. If required, the hubs are axially fixed by pins through the hub and the shaft.

7. Bushings

Intermediate bushings (Figure 3.44F), which are tapered, come in two basic configurations: with and without a flange. They come in many standard sizes of outside diameter with variable bore diameters. They are usually cheaper than coupling hubs and thus can be stocked in a wide range of sizes for use on different pieces of equipment. This type of interface connection is easy to assemble. It has a clearance fit at assembly, but when properly assembled it provides for an interference fit between hub and shaft. The use of tapered bushings tends to increase the initial cost of the coupling and sometimes increases the size of the coupling required. Standard sizes are available for bore sizes from approximately 1/2 to 10 in. Three of the most common types available are

1. The Dodge taper lock
2. The Eaton QD bushing
3. The Browning XT bushing

8. Splines

A spline (Figure 3.44G) is an interface connection consisting of integral keys and keyways equally spaced around a bore. In general, the teeth have a pressure angle of less than 45°. If the pressure angle is over 45°, it is usually called a serration rather than a spline.

a. Involute Splines. An involute spline is a spline with teeth having an involute profile. This form is the most common and has a relatively high torque capacity due to its configuration. Involute splines are economically manufactured by hobbing (external teeth) and shaping (external and internal teeth).

The various tooth proportions and tolerances are covered by a standard on involute splines and inspection [American National Standards Institute (ANSI) B92.1a]. The two most commonly used fits used for involute splines are

1. The major diameter fit
2. The side fit

In the major diameter fit, the mating diameter fit between the mating parts is tightly fitted at the major diameter of the gear teeth. The sides of the teeth have some clearance. This type of fit requires fitting parts or tight-tolerance parts, which tends to increase cost.

The side-fitted spline is most commonly used. The teeth contact only at the sides; the side clearance is minimal. The major and minor diameters have relatively large clearances. In high-torque and/or high-speed applications the side-fitted spline is used with locating pilots usually at the front and back of the hub. The pilots help prevent shifting, rocking, and fretting of the spline.

9. Keyless Fitted Hubs

The trend in high-speed and/or high-torque applications is toward the keyless taper-fitted hub connection. The first type of keyless hub connection provided high torque capacity but presented problems when the hub had to be removed. It usually had to be cut off, destroying the hub and sometimes the equipment shaft itself. Several hydraulic methods for the removal of shrunk-on parts were introduced by SKF and Siemens (Figure 3.44H) in the late 1940s and early 1950s. They provided a means of applying oil under pressure between the hub and the shaft so that the hub could be pressurized and then pulled off. With the Siemens method, the pressure against the chamfered shaft pushed the hub off the larger portion of the shaft.

a. Tapered Shaft. Tapered hydraulic shafts (Figure 3.44I) are becoming very popular. This is because the hubs cannot only be removed hydraulically but can also be mounted. This is an important advantage, particularly in the refinery and petrochemical industries, where open flames and other means of heating hubs to expand them may present a safety problem.

This type of interface connection requires more attention than the straight bore because it is easier to machine cylindrical bores and shafts than tapered ones. It is very important that the mating shaft and hub surfaces match. If the tapers do not match, the hubs can slip and/or produce high residual stresses that may cause the shaft to fail.

It is important that hubs on tapered shafts be properly advanced or pulled up on the shaft to provide the proper torque capacity of the interface. To determine the advance (pull-up) required to obtain a given interference, the following equation is used:

$$\text{Advance } (S) = 12\frac{i}{t}$$

where

i = diametral interference
t = taper (in./ft O.D. diameter)

If the taper is given in degrees,

$$t = \frac{\text{degrees}}{57.28} \times 12 = \text{in./ft}$$

The diametral interference can be found by multiplying the interference rate (in./in. shaft diameter) by the shaft diameter (D).

EXAMPLE

Given: 0.0015 in./in. shaft diameter; shaft diameter = 10 in.
Thus diametral interference $i = 0.015$ in.

Figure 3.52 gives the advance required for various tapers at different interferences.

As stated earlier, the contact between the bore of the hubs and the shaft is critical. The fit between the hub and the shaft should be checked by bluing. On hydraulic hubs, 70–80% contact is usually acceptable. If it is less than this, the part(s) may have to be remachined or lapped with special tools. The tapered shaft or hub should not be made without a plug and ring set so that parts can be checked with these master gages when they are being made. If this occurs, the likelihood of mismatched parts is minimized. When a mismatch is found, some people attempt to lap the hub to the shaft. This can be dangerous, as it can produce steps in the shafts and really makes the mismatched condition worse. Other people use the master gages for lapping the hub or shaft and end up destroying them so that the next time this set of gages is used, very odd looking contact patterns emerge.

When a mismatch occurs, the first thing to do is to check the plug gage with the ring gage for contact. If they do not fit properly, get a new set, because you have a problem. If the gages check out, check the hub

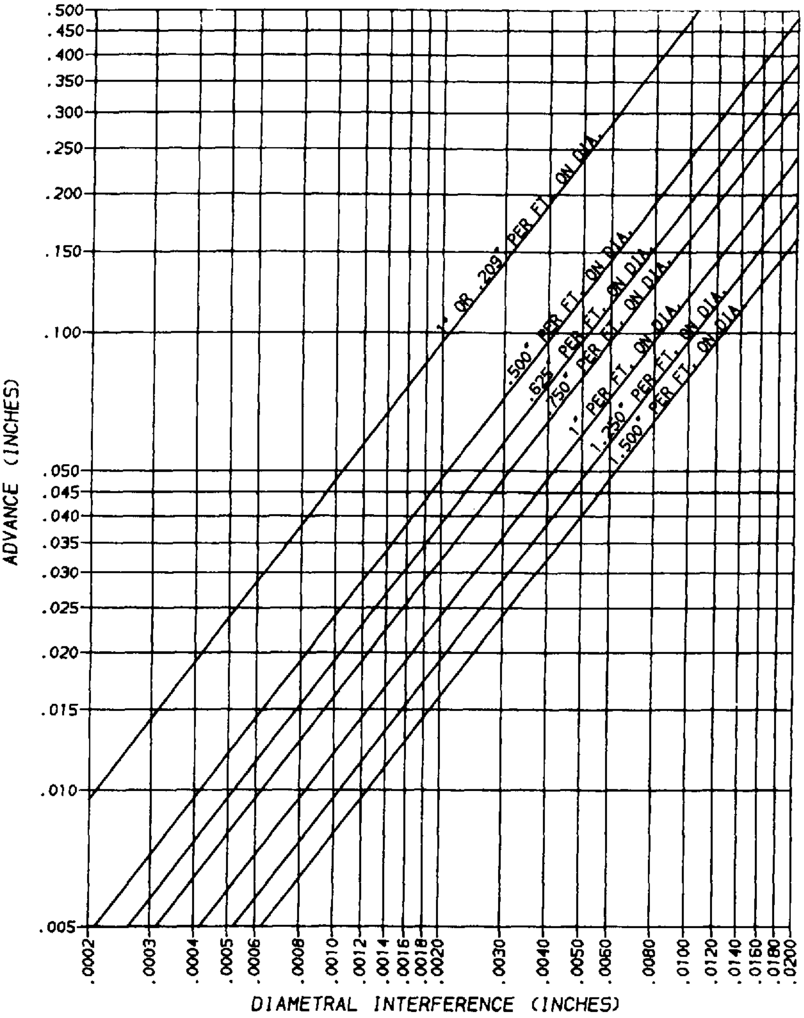


FIGURE 3.52 Amount of hub advance for various tapers.

and shaft with them. Once you find which part is wrong, have it fixed. Remachine (grind) it if possible, being careful to calculate a new advance so that any change does not cause problems with setup dimensions of the equipment or the coupling. If machining the parts is not possible, make a lapping plug or ring from the master gages. If none of the above work, make a ring gage to fit the shaft and then make a special plug gage so that a new coupling hub can be made to fit this special shaft. There are many design variations and preferences with regard to hydraulic hubs.

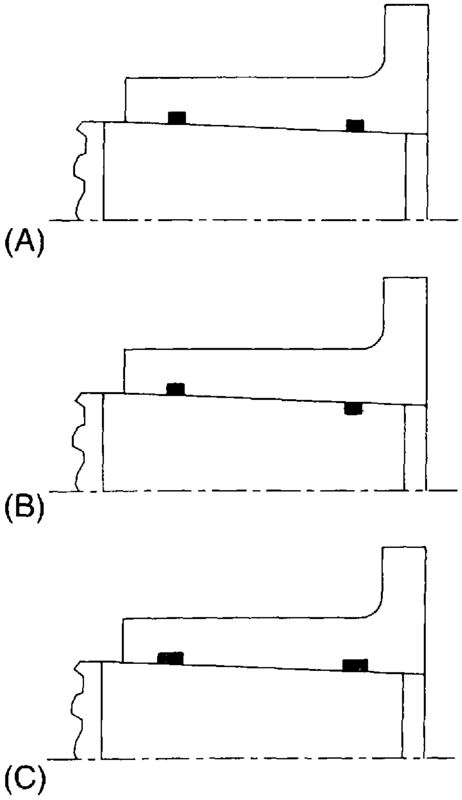


FIGURE 3.53 Various O-ring arrangements.

b. Sealing. Are O-rings required or not? O-rings (Figure 3.53) are required if the hub is to be hydraulically mounted and removed. If hubs are heat mounted and removed hydraulically, O-rings are not necessary. The hydraulic pressure causes nonuniform expansion of a hub, particularly at the ends of the hub, and this acts to prevent pressure loss and leakage so that the hub can be removed.

There are two common O-ring configurations (see Figures 3.53A and 3.53B); selection is a matter of preference. Most hydraulic hubs that use O-rings use backup rings (usually nylon or Teflon). The O-rings must be inboard of the hub, facing each other. The backup rings must be installed outboard on the ends of the hubs (Figure 3.53C).

c. Injection Holes. Oil must be injected between the hub and the shaft. The oil can be introduced through either the hub or the shaft (see Figure 3.54).

d. Stress Relieving. The stress distribution generated by the contact pressure is higher at the ends of the hubs and is considerably higher at the inboard end of hub. The easiest way to reduce these stresses is with a relief groove in the hub (Figure 3.55).

e. Hubs. Three types of hub designs are shown in Figure 3.56. Steps or flanges in hubs tend to present difficulties with hydraulic mounting or dismounting of hubs unless sufficient pressure is supplied to expand these areas. Flanges over shafts are difficult to avoid, particularly when reduced moment couplings are required. Large flanges should overhang the equipment shaft (see Figure 3.56B) so that the mounting pressure is reduced. If the face of the flange is directly over the end of the shaft, a stress-relief groove at the hub face may be required to help reduce stresses.

f. Retaining Mechanisms. Once a hub is installed, some type of axial retention (Figure 3.57) should be used for safety. An improperly installed hub could slip down the shaft and even off the shaft, causing serious damage to the equipment.

g. Hydraulic Mounting and Dismounting Equipment. Figure 3.58 shows a typical mechanism of which there are several variations. Most provide a means of injecting oil and a means of pushing the hub up on

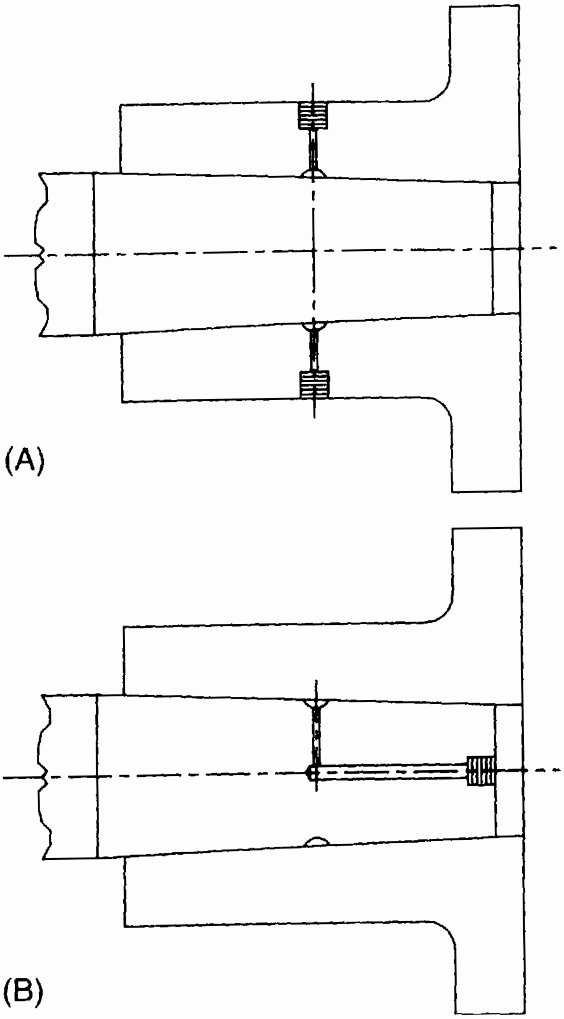


Figure 3.54 Types of oil injection methods. (A) Through the hub. (B) Through the shaft.

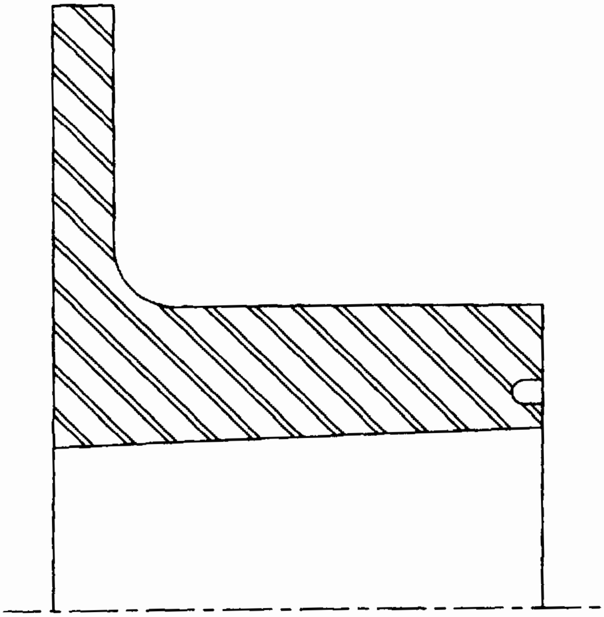


FIGURE 3.55 Stress-relief grooves for taper-bored hubs.

the shaft. Since in most cases the hydraulic pressures are between 20,000 and 30,000 psi, special high-pressure tubing and fittings are required.

B. Flanges

Flanges (see Figure 3.44) are very common interface connections on large-horsepower equipment (large steam turbines) and in applications to help reduce the overhung weight. It would be nice if there were only one flange connection, but there are so many different types that a standard would restrict the use of many of them.

Since there are so many different flange connections, it becomes important that exact specifications be supplied. In most cases, the equipment manufacturer's flange is fixed and the coupling must be made to bolt up to it, although in some cases the equipment flanges are made to accommodate a coupling. Whichever is the case, the proper information

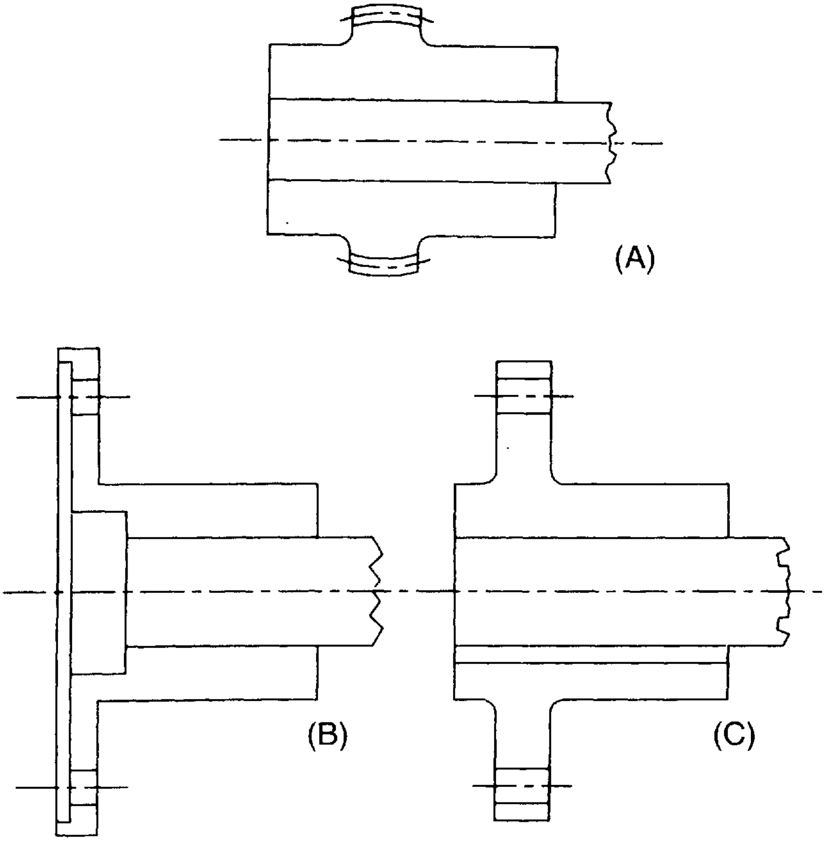


Figure 3.56 Various types of hub designs. (A) Flex hub. (B) Rigid flange overhanging shaft and. (C) Rigid hub flange over shaft.

has to be supplied; otherwise, the user will end up with a coupling that will not bolt up to the equipment.

1. Shrouded Versus Exposed Bolts

In the past, shrouded bolt couplings were used to protect people, but today all couplings have some type of guard. The shrouded coupling can in some instances reduce windage and noise, depending on the design of the coupling guard. The shrouded flange coupling is usually heavier than the exposed flange coupling. Exposed bolt couplings are usually

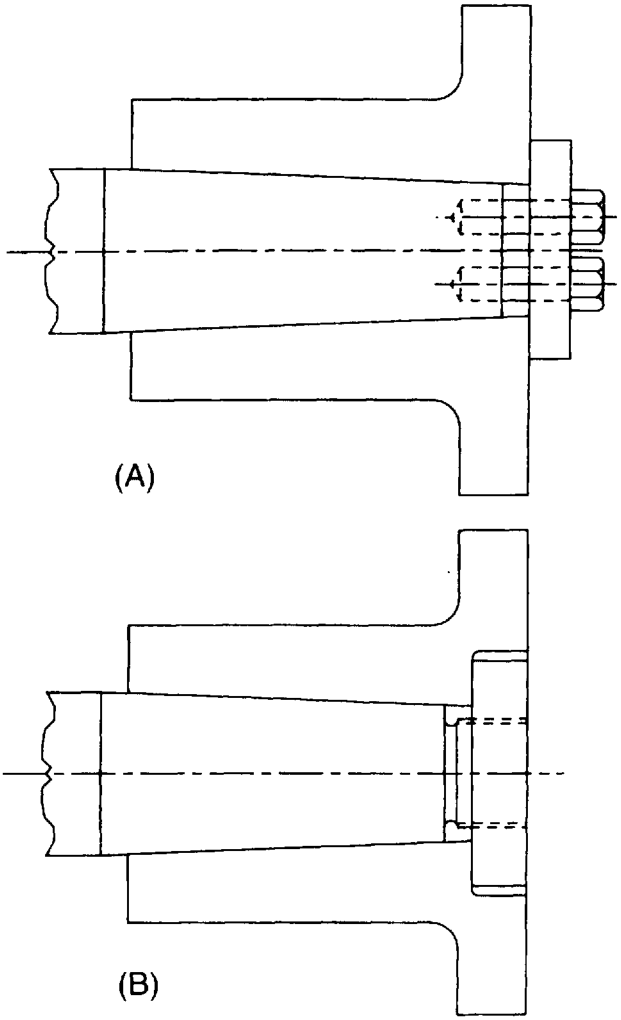


FIGURE 3.57 Types of hub retention. (A) Bolted plate. (B) Thread nut.

cheaper. The noise level can be higher, but if the coupling guard is designed properly, the difference between the two types is usually insignificant compared to the rest of the system. In fact, on applications where the coupling guard is fitted tightly around the coupling, an exposed bolt coupling may produce less windage and horsepower loss

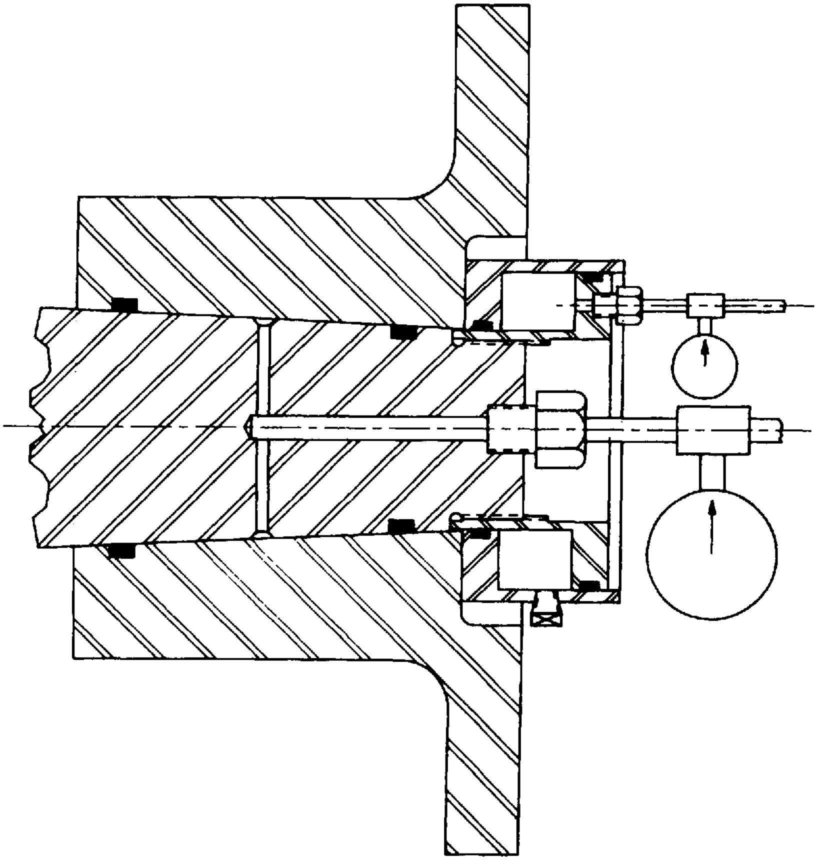


FIGURE 3.58 Hydraulic hub mounting equipment.

than those produced by a shrouded coupling. This is true simply because the flange length of the shrouded bolt coupling is greater than that of the exposed bolt coupling.

2. Face Keys and Face Splines

Face keys (Figure 3.44K) and face splines (Figure 3.44L) are used on high-torque applications where bolts constantly loosen and break. When face keys are incorporated into a flange, care must be exercised so that the flanges are not weakened from the keyway. Usually, the keyway depths are approximately one-half of the flange thickness. Also important are keyway radii and key-to-keyway fit. Face splines provide even higher torque capacity than face keys and are relatively expensive.

Both face keys and face splines are used with heavy-duty drives such as universal joints and are also used where a high-capacity connection must be disassembled with ease. The addition of face keys or splines allows for a reduction in the number of bolts used with the flange connection.

C. Summary

Interface connections are important and should be considered carefully by the coupling selector. The coupling manufacturer should be supplied with specifications detailing all aspects of the interface. This will prevent delays, misunderstandings, and possibly failure of the coupling and/or the equipment.

This page intentionally left blank

4

Installation and Maintenance

I. INTRODUCTION

In this chapter we discuss the installation and maintenance of the various types of couplings available on the market. Most coupling manufacturers provide information and recommendations relevant to the installation, maintenance, and lubrication of their couplings. If this information is not included with a coupling, the coupling manufacturer should be contacted before installation is begun. A good maintenance department has a file where this information is kept for reference.

So why should you read this chapter? Because it presents a discussion of general procedures and some of the reasons for these practices and provides a good starting place for the person who has not installed a coupling before.

Parts of Section III. A, I, and K and some of the illustrations in these sections are taken from *Shaft Alignment Handbook*, 2nd ed., by John Piotrowski (Marcel Dekker, 1995).

Parts of this chapter and some of the illustrations on installation are taken from *Flexible Couplings Installation* by Michael Calistrat (a Kop-Flex paper, 1968).

II. INSTALLATION OF COUPLINGS

A. Preparation

Upon receiving a coupling, proceed as follows:

1. Obtain a drawing (if one exists) of the coupling and a copy of the coupling manufacturer's and/or equipment builder's installation and maintenance manual.
2. Review and read the information. If any questions arise, contact the coupling manufacturer or equipment builder.
3. Inspect the components to be sure that all parts that were ordered have been received.
4. If the coupling is going to be stored for a period of time, make sure that the parts are properly protected. Steel parts are usually coated with an oil or wax coating. Rubber parts are usually wrapped or packaged so that exposure to light and air is minimized (light and air tend to harden rubber and some elastomers). If the coupling is to be stored for more than a year, check with the coupling manufacturer for special instructions. Particular attention should be directed to seals, rubber elements, and any prelubricated parts when long-term storage is anticipated.
5. Prior to assembling the coupling, clean and disassemble the components. *Note:* Some subassemblies should not be taken apart. Check specific coupling instructions to determine what to disassemble. *This is very important.*
6. Check for burrs and nicks on the mating surfaces. If they exist, remove them (stone, file, sand).
7. Measure and inspect
   a. Bore dimensions
   b. Keyways
   c. Coupling lengths
   d. Distance between shaft ends (DBSE)
   e. Diameter of bolt circle (DBC), bolts, and holes of the mating flange
   f. Any other dimensions

   to ensure that the coupling will mate properly with the equipment.

8. Obtain hardware, keys, tools, and anything else not supplied but needed to complete the assembly.

B. Hub Installation

Normally, hubs are mounted before the equipment is aligned. In some instances, the hubs may be mounted by the equipment manufacturer. Described below are steps in the installation procedure for various fittings.

1. Shrink Fit vs. Clearance Fit

Questions often arise as to when to use a clearance fit as opposed to an interference fit and how much shrink should one use. It is very important to ensure the proper fit of a hub on its shaft.

a. Clearance Fit (with Keys and Setscrews). Generally a clearance fit with setscrews is used on applications under 10 hp and shafts up to 2 in. in diameter. Even though torque is transmitted through the keys, setscrews are usually provided over the key to minimize the play and resist the moments and forces resulting from misalignment and unbalance.

All flexible couplings resist misalignment with reactionary moments and forces. The magnitude of this resistance depends on the type and size of the coupling. These moments and forces are caused by friction in lubricated couplings and the flexing of material in nonlubricated couplings. Such moments and forces can cause a loose hub to rock on its shaft. Fretting can occur, which can cause failure of the shaft and/or the hub. Therefore, clearance fit applications should be limited to relatively small horsepower applications where the reactionary loads are usually low (10 hp).

b. Shrink Fit. For applications up to 1000 hp, the use of keys with a shrink fit that conforms to AGMA recommendations is usually sufficient to keep the hub from rocking on the shaft. For standard couplings with keys, the shrink should not exceed 0.00075 in./in. Caution must be exercised when specifying heavy shrinks for keyed hubs. Remember that on keyed hubs the purpose of the shrink is to keep the hub axially positioned on the shaft and resist the moments and forces generated by unbalance and misalignment. If one tries (or wants) to drive totally through the shrink fit when a key is used, this can lead to hubs splitting over the keyways. If this is what you want, then take the key out and go to a heavier interference. Some standard couplings have hubs made from material that cannot handle shrink fit or can handle very little shrink.

For high-horsepower applications and high-speed applications (API 671 and ISO), keyless fits (straight and tapered) are commonly used. The shrink for keyless fits needs to be sufficient to handle the expected normal and transient loads, with an interference range from 0.0015 in./in. to as high as 0.003 in./in.

Table 1 shows the maximum interfaces that can generally be used with various hubs with and without keys. It is suggested that you consult with the coupling manufacturer to find out what maximum shrink they recommend for their hubs.

2. Straight Shafts

a. Clearance Fit. This type of installation is relatively simple.

1. Rotate the shaft to bring the keyway up, and install the key(s) in the shaft keyway(s).
2. Coat the shaft and key with a thin layer of antiseize compound.
3. Make sure that any part that will not slide over the coupling hub is placed back on the shaft. This includes seals, carriers, covers, and, on gear couplings, the sleeves.

TABLE 1

Material hardness (BHN)	Amount of shrink	Key
110	0.0003 in./in.	Yes
160	0.0005 in./in.	Yes
250	0.00075 in./in.	Yes
160	0.001 in./in.	No
250	0.00175 in./in.	No
300	0.0025 in./in.	No
330	0.003 in./in.	No

4. Push the hub onto the shaft until the face of the hub is flush with the end of the shaft. (*Note:* Some coupling hubs are not mounted flush. Check specific instructions.) If the hub does not slide onto the shaft, check the clearances between the bore and the shaft. In addition, check to be sure that there is clearance between the keys at the sides of the coupling keyway and above the key.
5. Lock the hub in position (usually with setscrews). Make sure that setscrews have a locking feature such as a Nylok insert, or use locking compound. Some hubs use bolts, nuts, or other means to secure the hub in place. See the specific instructions.

b. Interference Fit. This type of installation is the same as that for a clearance fit, with the exception that the hubs must be heated before they slide onto the shaft. The coupling manufacturer usually supplies information as to how to heat the hub and to what temperature. For steel hubs, 160°F is required for every 0.001 in. of interference per inch of hub diameter (0.001 in./in.). For example, for a steel hub with a 4 in. bore with an interference of 0.003, the required temperature is

$$\frac{0.00075}{0.001} \times 160°\text{F} = 120°\text{F}$$

Therefore, if the shaft temperature is 80°F, the hub temperature must be 200°F. This does not account for human factors such as cooling due to handling time, errors in measurement, and so on. As a general rule, add 50–75°F to the calculated expansion temperatures to account for these factors. The hub should be heated in an oil bath or an oven. A torch or open flame should not be used because it could cause localized distortion or softening of the hub material; it could also cause an explosion in some atmospheres. Oil bath heating is usually limited to approximately 350°F, or under the flash point of the oil used. Special handling devices are required: tongs, threaded rods placed in taped holes in the hub, and so on. Oven heating and the use of induction heaters offer some advantages over heating in an oil bath. Parts can be heated to higher temperatures (usually not exceeding 600°F) and can be handled with heat-resistant gloves. In any event, extreme care must be exercised when handling heated hubs to avoid injury to personnel.

It is also important when mounting interference fit hubs to make sure that clearance exists over the top of keys. Otherwise, when the hub cools, it will rest on the key and produce high stresses In the hub that could cause it to fail.

3. Straight Shafts with Intermediate Bushings or Locking Rings

Intermediate bushings (see Figures 4.1A–C) come in two basic configurations: internal or external. Although installation varies with bushing design, the net result is an interference fit between the hub and the shaft. Tightening axial screws to draw up opposing tapered rings develops the required interference. Generally, to assemble: (1) Insert the bushing into the hub without tightening the screws or bolts. Then (2) slide the hub and bushing onto the shaft. Since the bushing is tapered, (3) tighten the screws or the shaft. Once the hub is at the correct position, (4) tighten the screws gradually in a crisscross pattern to the specific torque. Bolts are tightened on a coupling similar to the way in which lugs are tightened on the wheel of an automobile.

To facilitate the installation and tightening of the bushing, all parts, including the bolts or the setscrews, should be oiled. *Do not* use grease, as grease might prevent the proper tightening of the bushing. Refer to the specific instructions for further recommendations and the correct torquing value.

Tapered bushings have two advantages over straight shafts:

1. They slide easily onto the shaft, but once drawn up they provide for an interference fit. The interference is usually not sufficient to transmit full operating torque, but the bushings incorporate integral keys or have keyways.
2. They come in standard sizes but have various standard bores for each size. Stocking a few hubs bored for a tapered bushing is usually an economical way to set up stock for a large plant, because they can be used on a variety of bores.

There are two main disadvantages of tapered bushings:

1. They increase the total cost of the coupling.
2. They can cause hub failures if they are improperly installed (overtightened).

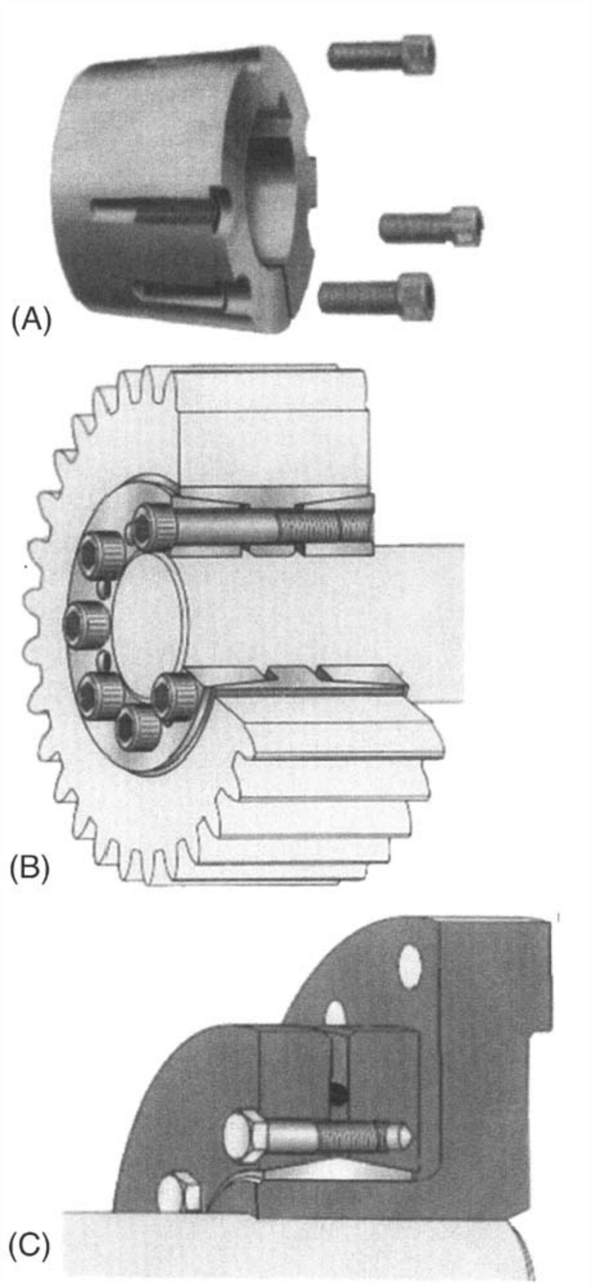


Figure 4.1 (A), (B) Intermediate bushings. (C) Locking ring.

4. Tapered Shafts

Tapered shafts have the advantage that the interference between the hub and the shaft can be accomplished by advancing the hub on the shaft. Depending on the amount of interference, the hub may be drawn up with nuts or by heating. A hub can usually be removed more easily from a tapered shaft than from a straight shaft.

Note: The shallow tapers used in coupling applications are usually self-locking. This means that the friction forces resisting the hub advance are greater than the forces pushing the hub onto the shaft. Therefore, the only way to accomplish the required advance is to heat the hub before installation.

Applications using tapered bores require more attention than those using straight shafts because it is easier to machine two cylindrical surfaces that match than to machine two tapered surfaces that match. The hub can be overstressed if it is advanced too far on the shaft. Dirt and surface imperfections can restrain the hub advance and give the false impression that the desired interference has been reached.

To determine the draw-up required to obtain the desired interference, use the equation

$$\text{Draw-up (in.)} = 12\,\frac{i}{T}$$

where

i = diametral interference (in.)
T = taper (in./ft in diameter)

The area of contact between the bore of the hub and the shaft should be checked with machinist's bluing. Contact of 50–80% is in the range of acceptability; usually, 70% is the most desirable. If less than the required contact is achieved, the contact can be increased by lapping the bore and/or the shaft with a plug or ring made from a master plug and ring gage. It is not recommended that the master gages or shaft be used to lap the hub, as the gages could end up with ridges. Ridges in the hub or shaft will prevent proper hub installation and could cause the hub or shaft to fail because of stress concentrations.

a. Installation. The method used to install a hub on a tapered shaft depends on the degree of interference.

Light interference (under 0.0005 in./in.). When the interference is less than 0.0005 in./in., the hub can usually be advanced without heating. Although heating the hub is the most common method, the hub can usually be advanced by tightening the retaining nut or plate on the shaft. It is also common practice when light interference is used with a combination of keys and a retaining nut or plate to use a light grease or antisieze compound between the hub, shaft, and threads on the shaft and nut. This should help facilitate installation and future removal and help prevent shaft and/or bore gauling.

Medium interference (usually 0.0005–0.0015 in./in.). When the interference is over 0.0005 in./in., the force required to advance the hub could become too great for manual assembly. When this occurs, the hub *must* be heat-mounted or hydraulically mounted. Heating hubs for mounting is the most common method. Regardless of the method used, the amount of draw-up must be measured.

Heavy interference (usually over 0.0015 in./in.). When the interference is over 0.0015 in./in., hubs are usually heat-mounted and removed hydraulically. Some users prefer to both mount and remove hubs hydraulically.

b. General Installation Guidelines. The following is recommended as a general guide when installing hubs on an equipment shaft.

1. Install the hub on the shaft, being careful that the parts mate properly and are burr-free and clean. Using a depth gage or dial indicator (see Figure 4.2A), measure and record the initial reading.
2. If hydraulic assist is to be used, remove the hub and lubricate the bore or shaft; if not, heat the hub in oil or in an oven. When using a heating method for mounting hubs, it is best to provide a positive step, such as a clamp on the shaft, to ensure proper draw-up (see Figure 4.2B). The reason for this is that a hub advanced too far may not be removable (too much force required or not enough hydraulic pressure available to remove the hub) and normally requires that the hubs be cut off.
3. Install the hub and advance it the required amount.

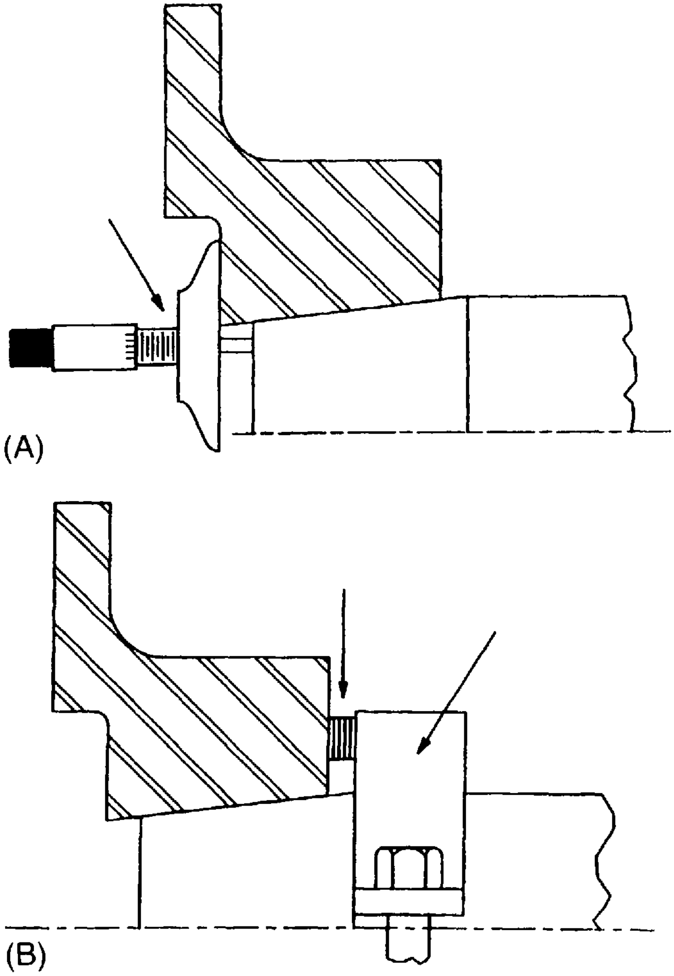


Figure 4.2 (A), (B) Positioning hubs.

4. Properly tighten the shaft nut and lock it in place. Locking is done with a tab washer or setscrew.

5. Rough and Semifinished Bores

Most coupling manufacturers will supply couplings with rough or semifinished bores. This is the usual practice with spare couplings so that as equipment shafts are remachined, the spare coupling can be properly fitted. It also helps to reduce inventory requirements.

It is important that the user properly bore and key these couplings; otherwise, the interface torque-transmission capabilities can be reduced or the coupling balance (or unbalance) can be upset.

Recommendations should be obtained from the specific coupling manufacturer. As a general guide for straight bores, the hub must be placed in a lathe so that it is perpendicular to and concentric with its controlling diameters. On rigid hubs the pilot and face are usually the controlling diameter and surface that should be used to bore (see Figure 4.3A). On flexible hubs (gear and chain), the gear major diameter (O.D.) and hub face act as the controlling diameter and surface. [*Note:* Some manufacturers use the hub barrel as the controlling diameter (see Figure 4.3B).]

Some coupling manufacturers supply semifinished bore couplings. In this case, the finished bore should be machined using the semifinished bore as the controlling diameter. Indicate the bore-in for concentricity and straightness.

There are three types of errors that can occur when hubs are bored:

1. The bore diameter is incorrect. Too much interference will cause installation problems and may damage the hub or shaft, whereas too much clearance can produce unbalance forces that may be unacceptable to the system.
2. The bore is eccentric but is parallel to the hub axis. This can produce unbalance forces that maybe unacceptable to the system (see Figure 4.4).
3. The bore is at an angle to the O.D. of the hub. In this case the misalignment capacity of the coupling is reduced (see Figure 4.5).

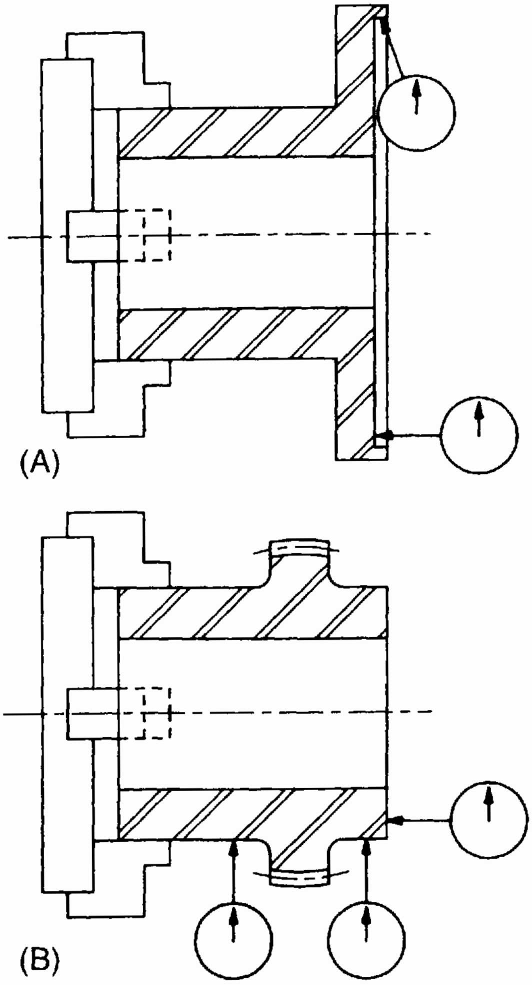


Figure 4.3 (A), (B) Setup for reboring straight shaft hubs.

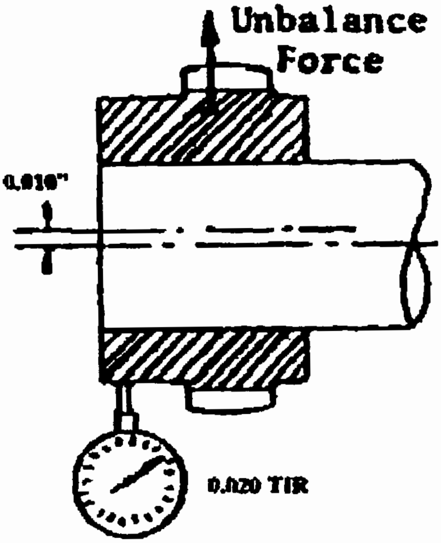


FIGURE 4.4 Eccentric hub bore.

6. Key Fit

Most couplings must have one or two keyways cut in the hub. These should be cut according to the tolerances listed in AGMA 9002-A86 (see Section IV of Chapter 3). Particular attention should be given to the following:

1. Keyway offset (The centerline of the keyways must not intersect with the centerline of the bore.)
2. Keyway parallelism
3. Keyway lead
4. Keyway width and height

The fitting of keys is important to ensure the proper capacity of the interface. Refer to Section IV of Chapter 3 and the AGMA standards on keyways and keys. As a general rule, three fits must be checked:

1. The key should fit tightly in the shaft keyways.

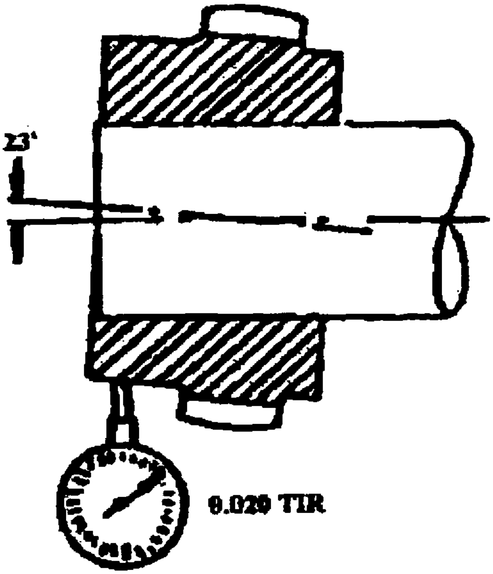


Figure 4.5 Skewed hub bores.

2. The key should have a sliding fit (but not be too loose) in the hub keyway.
3. The key should have a clearance fit radial with the hub keyway at the top of the key (see Figure 4.6A).

In addition, the key should have chamfered corners so that it fits without riding on the keyway radii (see Figure 4.6B). A sloppily fitted key can cause the keys to roll or shear when loaded. The results of a sloppy key fit are shown in Figure 4.6C. The forces generated by torque produce a moment, and this moment tends to roll the key and can cause very high loading at the key edges. On the other hand, too tight a fit will make assembly very difficult and increase the residual stresses, which could cause premature failure of the hub and/or shaft.

A key that is too high in the keyway could cause the hub to split (see Figure 4.6D). When there is too much clearance at the top or sides of a key, a path is provided for lubricant to squeeze out. For lubricated

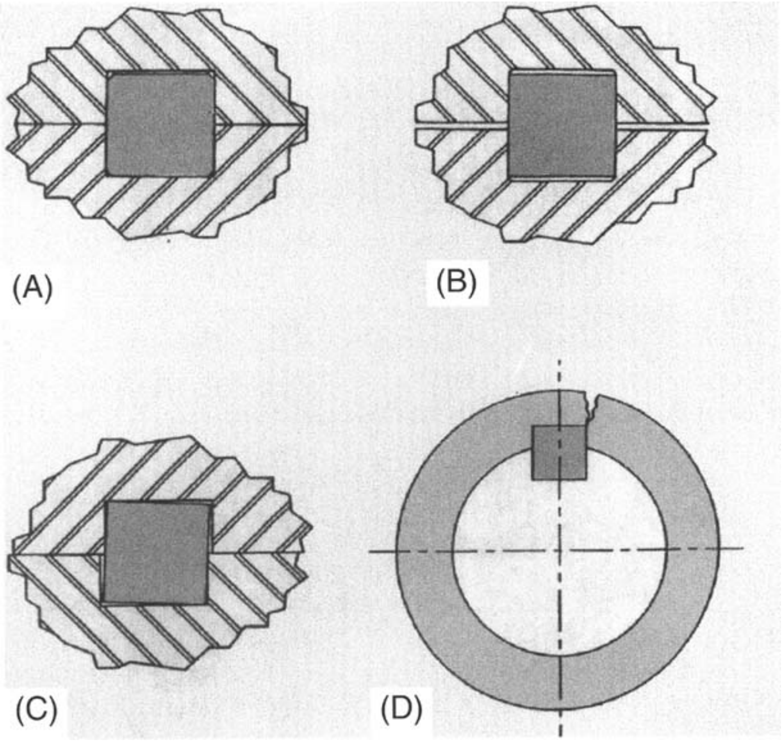


FIGURE 4.6 Improperly fitted keys.

couplings, clearances between keys must be sealed to prevent loss of lubricant and thus starvation of the coupling.

7. Shrink Fit Effect on Pilots

When a coupling hub is pressed onto a shaft (shrink fitted), the bore expands to fit the larger shaft. Not only does the bore grow, but so does any diameter directly over the bore. Therefore if the coupling has piloting fits that are directly over the shaft, they too will grow.

$$G = \frac{D_B}{D_H} i$$

where

G = growth of pilot (in.)
D_B = bore Diameter (in.)
D_H = pilot O.D. (in.)
i = amount of shrink (in.)

Compensation for this growth must be accounted for. Outside rabbets can grow so the assembly becomes difficult or impossible. Inside rabbets may loosen and cause unbalance problems. In the case of gear couplings, the tooth tips may grow so they will no longer slide in the sleeve; this causes the coupling to be mechanically locked. Coupling manufacturers account for this in their designs, but if one elects to buy rough-bored couplings and decides to use a heavy shrink fit, the growth of the hub tooth might be greater than allowed by design, and "lockup" may occur.

C. Installation Considerations (Do's and Don'ts)

Most coupling failures occur relatively soon after installation or after the coupling has been taken apart and reassembled on the equipment. Why? Because somebody didn't follow the installation instructions supplied. They have been installing couplings for years, and who needs to read those boring instructions? Let me tell you. **Read the instructions!** If you do not have instructions, *get them!* If you follow them, you will have a very high probability of success.

1. Things To Do—A Sure Way To Success

Consider total cost when selecting couplings. Look at initial cost versus total operating cost.
Provide the coupling manufacturer with as much information as is justified by the criticality of the equipment.
If you are changing out types, do not do it just for the sake of change.
Be sure that maintenance people are properly trained in the installation and maintenance of the type of coupling being used.
Use the bolts supplied. If one gets lost, buy a spare set from your coupling supplier.
Use a torque wrench to torque bolts.
Align equipment to limits specified by the equipment manufac-

turer, and recheck whenever major maintenance is done on the equipment.

Lubricate as required, with the right lubricant and at the proper intervals.

2. Things Not To Do—A Sure Way To Failure

The hub will not fit on the shaft. Get out the torch and heat the hub, or open up the bore until it slides on.

The keys are too tight. Grind them until they fall in.

You lost a bolt. Get one out of the bolt can in the maintenance department.

Decide that the bolts are snug enough as they are.

The pump or motor is out of balance. Take it all out of (or on) the coupling.

A part of the coupling looks bad. Replace it with a part you removed from another coupling last year.

Put grease in the coupling—after all, it works well in the bearings, why not in the coupling?

Check alignment? Why? It ran okay for two years with no problems.

Ask for a coupling to fit on a 3.0 in. shaft with a 1/2 in. keyway in it. Give no other information for the replacement coupling.

Buy a different type of coupling because it is 30% less expensive than what you've been using.

III. ALIGNMENT OF EQUIPMENT

Once the hubs have been assembled, the next step is usually to align the equipment, although some people prefer to assemble the coupling and then check and align the equipment with the coupling fully assembled. The danger here is that if the alignment is too far off, the static misalignment imposed on the coupling while it is being assembled may damage it, which could result in premature failure. Therefore, it is suggested that at a minimum, the equipment should be rough-aligned before the coupling is installed.

Types of alignment and the steps required are covered in general in the following discussion. The subject of equipment alignment could fill a book by itself and is a very common subject of debate. Alignment tools vary from the simple straightedge to very sophisticated laser methods.

You may be wondering why we need to worry about alignment, since flexible couplings are designed to allow some flexibility in alignment. One important reason is that the coupling's useful life depends on the amount of misalignment. The other is that the equipment usually needs to be aligned much better than the coupling's capacity in order to run properly. If it were possible to align equipment perfectly and we could be sure that it would stay aligned, a flexible coupling would not be required. A third reason for accurate initial alignment is that alignment changes over time due to bearing wear, foundation settling, and thermal changes. When alignment is done carefully to begin with (see Figure 4.7), there should be enough capacity left in the coupling for it to handle any foreseeable increases in misalignment.

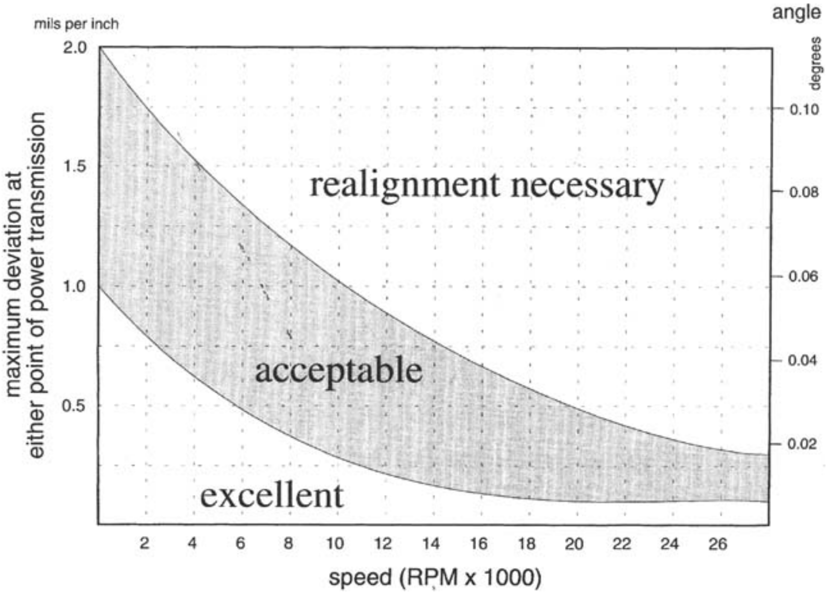


FIGURE 4.7 Misalignment tolerance guide.

A. The Importance of Alignment

Misaligned rotating machinery has caused, and will continue to cause, heavy financial losses in almost every industry. It is difficult to calculate how much money is wasted on prematurely damaged machinery, lost production, and excessive energy consumption due to misalignment. If you just consider the number of pieces of rotating machinery there may be, say, within a 50 mile radius, you can get a better idea of the magnitude of the potential loss in downtime and maintenance cost because the equipment went out of alignment.

The capability of having most of your rotating equipment well aligned and running smoothly is directly related to your knowledge, ability, and desire to do the job properly. It seems foolish to install well-designed or rebuilt machinery only to watch it be destroyed in 1 or 2 years (or even more quickly) because no one wanted to spend 4–8 hours carefully attaching rotating machinery to frames and foundations, aligning the shafts, and monitoring the alignment position over time to allow the equipment to run for 3–6 years (and even longer) instead.

Figure 4.8 lists the three key items that must be known to attain accurate and precise alignment.

When aligning rotating machinery there are 3 things you need to know

1. **Where is the machinery at when the equipment is not running?**
2. **What position will the machinery move to when operating?**
3. **If the machinery moves from off-line to running conditions, what acceptable range of positions could the machinery shafts be in when the machinery is aligned off line and still maintain acceptable alignment tolerances during operation?**

or simply . . .

Where are they?
Where will they go?
Where should they be?

FIGURE 4.8 Key items to attain accurate alignment.

Most rotating machinery operates with some misalignment. Perfect alignment is nearly impossible to achieve in the real world. In fact, a small amount of misalignment is really not that bad. Gear-type couplings and shafts with universal joint drives, for example, must have some misalignment in order to maintain lubrication at the points of power transmission during rotation. So whenever we deal with alignment, it is important to know when to stop moving the machinery. There comes a point where no additional beneficial returns can be achieved if alignment tolerances have been obtained.

Few people who are responsible for aligning machinery realize that machinery moves from off-line to running conditions.

B. Symptoms of Excessive Misalignment

Misalignment usually causes the following symptoms in rotating equipment:

1. Premature bearing, seal, shaft, or coupling failures.
2. Excessive radial and axial vibration. (*Note:* Different couplings exhibit different types of vibration behavior. It appears that the vibration is caused by the mechanical action that occurs in the coupling as it rotates. See Section III. K.)
3. High casing temperatures at or near the bearings or high discharge oil temperatures.
4. Excessive amounts of oil leakage at the bearing seals.
5. Loose foundation bolts.
6. Loose or broken coupling bolts.
7. Heating of the coupling while it is running and immediately after the unit is shut down. If the coupling is an elastomeric type, look for rubber powder inside the coupling shroud.
8. Unusually high number of coupling failures or unusually fast wear.
9. Breaking (or cracking) of the shafts at or close to the inboard bearings or coupling hubs.
10. Excessive amounts of grease (or oil) on the inside of the coupling guard.

C. Definition of Misalignment

Misalignment is the deviation of relative shaft position from a collinear axis of rotation measured at the points of power transmission when the equipment is running at normal operating conditions (see Figure 4.9). General-purpose equipment should be aligned to within 0.001 in. per inch of separation between flex points. For high-speed applications (generally over 3600 rpm), alignment should be within 0.0005 in. per inch of separation between flex points.

D. Types of Misalignment

Two rotating shafts can have the combinations of position shown in Figure 4.10, although conditions A and C are unlikely to occur in real life. A flexible coupling is usually employed as a double flex (except for elastomeric couplings). A gear-type coupling incorporating a spacer is

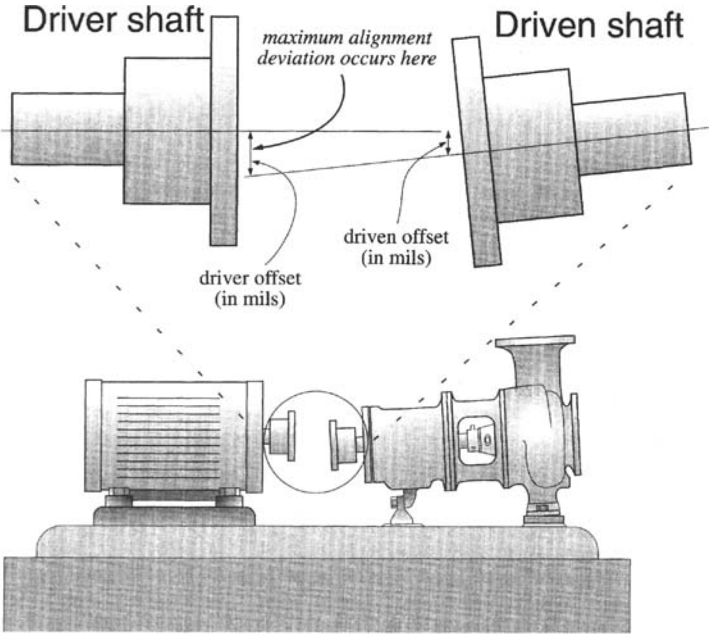


FIGURE 4.9 Definition of shaft misalignment.

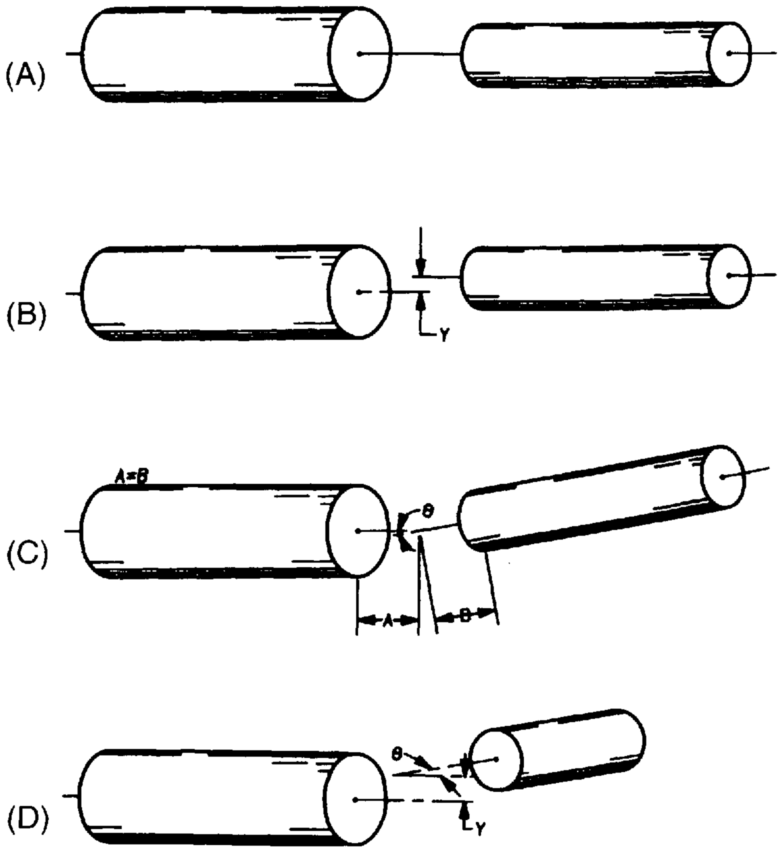


Figure 4.10 Shaft alignment conditions.

shown in Figure 4.11. Each coupling end (flex element) will accommodate only angular misalignment α_1 at one end and α_2 at the other; a double flex coupling can accommodate $\alpha = \alpha_1 + \alpha_2$ angular misalignment. It is important to remember that *there must be two flexible elements* to accommodate any offset misalignment (this does not apply to elastomeric couplings). The amount of offset misalignment a double flex coupling can accommodate is determined as follows:

$$\text{Offset } (S) = L \tan \alpha$$

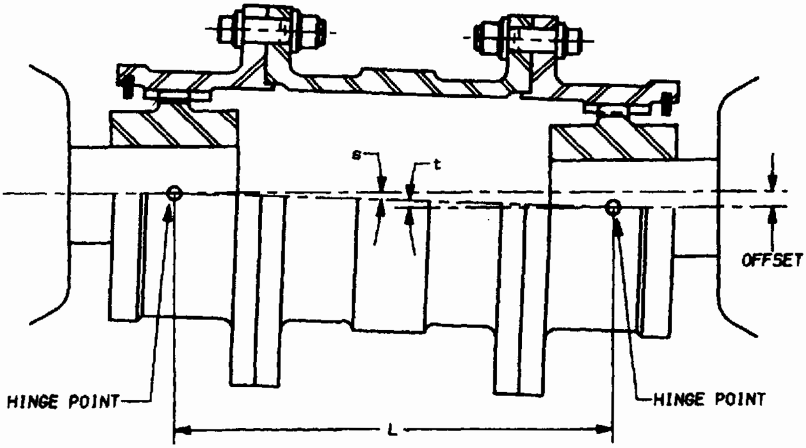


Figure 4.11 Misalignment of a spacer-type gear coupling.

where

α = angular misalignment (deg)
L = center distance between flex points

EXAMPLE 1

If a gear coupling is rated at $\alpha_1 = 1\ 1/2°$ per mesh and the distance between flex elements (flex points) is 6 in., the amount of offset capacity is

$$6 \tan 1.5° = 0.157 \text{ in.}$$

A flexible coupling can accommodate more offset if it is used in conjunction with a spacer. The longer the spacer, the greater the distance between flex points and the greater the allowable offset misalignment allowed.

E. The Difference Between Equipment Alignment and Coupling Tolerances

It is important to remember when selecting and/or using a coupling not to be misled by the term "allowable misalignment." In coupling catalogs, this is the rated capacity of the coupling and is not to be confused with the allowable limits set for successful operation of the equipment. Sometimes these numbers can be 10 times what the equipment can accept. The limits given by the coupling manufacturer are usually based on the life or fatigue limit of the various components of the coupling. Figure 4.7 shows misalignment tolerances that usually ensure successful operation of the system (including the coupling).

F. Straightedge, Feeler Gage, and Caliper Methods of Measuring Misalignment

The simplest way to align a coupling is to use a straightedge and a set of feeler gages and/or calipers (see Figure 4.12).

1. Offset Misalignment

Offset misalignment (Figures 4.12A and 4.12C) is measured by aligning a straightedge on one hub (using the same diameters) and measuring the gap between the straightedge and the other hub. Usually, the gap can be measured with a set of feeler gages. Remember that the angular misalignment created by this offset is

$$\alpha = \tan^{-1}\left(\frac{S}{L}\right)$$

EXAMPLE 2

If L is 6 in. and $S = 0.020$ in., then

$$\alpha = \tan^{-1}\frac{0.020}{6} = 0.19^\circ$$

2. Angular Misalignment

Angular misalignment is measured by taking measurements at maximum openings (between hubs) and a minimum opening (180°) directly with a feeler gage on close-coupled couplings (Figure 4.12A) and with a set of calipers on couplings where there is some distance between hubs. The formula is

$$\tan\alpha = \frac{t_1 - t_2}{D}$$

where

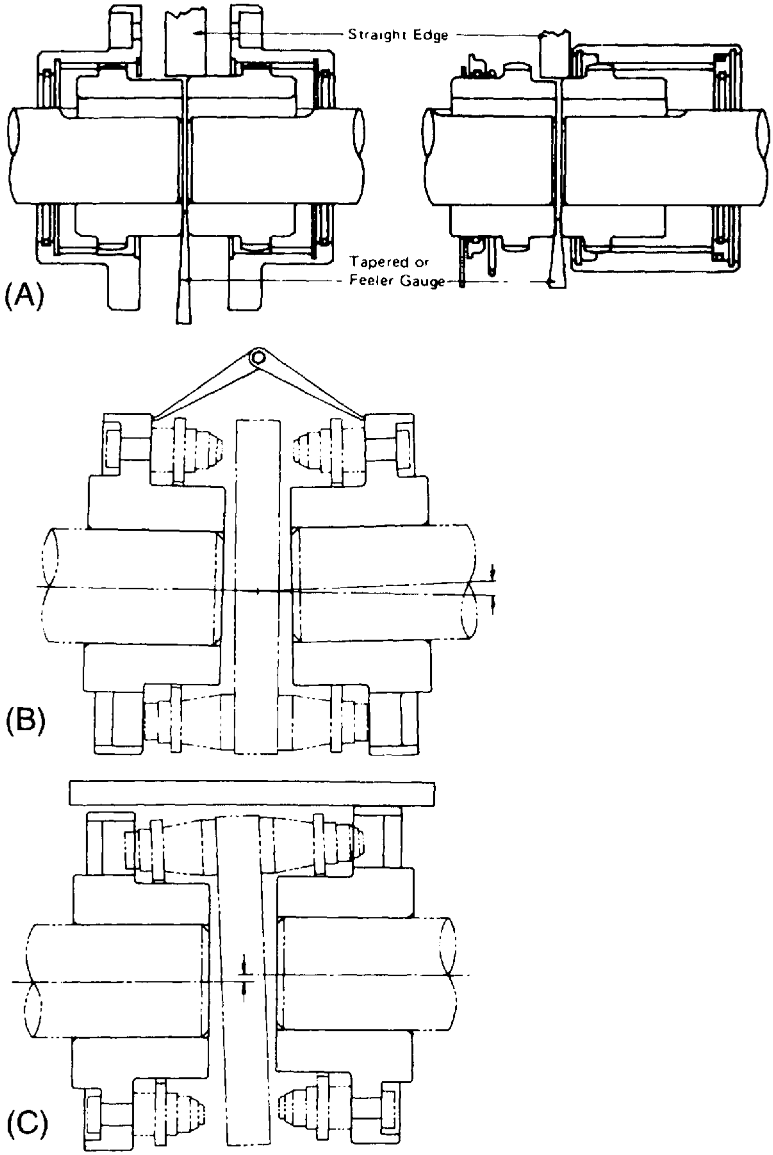


FIGURE 4.12 Alignment methods.

D = diameter at which readings are taken (in.)
t_1 = top gap reading (in.)
t_2 = bottom gap reading (in.)

EXAMPLE 3

If $t_1 = 0.250$ in., $t_2 = 0.225$ in., and $D = 4$ in., then

$$\tan\alpha = \frac{0.250 - 0.225}{4}$$

and

$$\alpha = 0.358^\circ$$

G. Dial Indicator Method

The most common method for aligning a flexible coupling is to use a dial indicator, ruler, vernier, or micrometer. The dial indicator(s) (see Figure 4.13) must be mounted on one or both hubs (clamped, magnetically held, or bolted). Both the clamping means and rigidity of setup must be considered. A setup that is too thin or flimsy could give false readings.

To obtain an accurate picture of misalignment, two readings must be taken: one with the indicator attached to the driving shaft and another with the indicator attached to the driven shaft. This procedure is called the *reverse indicator method.*

Figure 4.13A shows why a single reading can be misleading. Reading 1 makes it appear that there is perfect alignment, whereas reading 2 shows that there is an angular misalignment. Figure 4.13B shows that when two readings are the same but of opposite sign, only parallel misalignment exists. [*Note:* This offset misalignment is equal to one-half the total indicator reading (TIR).] Figure 4.13C shows that when two shafts have both offset and angular misalignment, only the angular misalignment can be calculated.

Figure 4.14 shows a method of measuring misalignment when there are large distances between pieces of equipment. This method gives the angular misalignment at each flexible element.

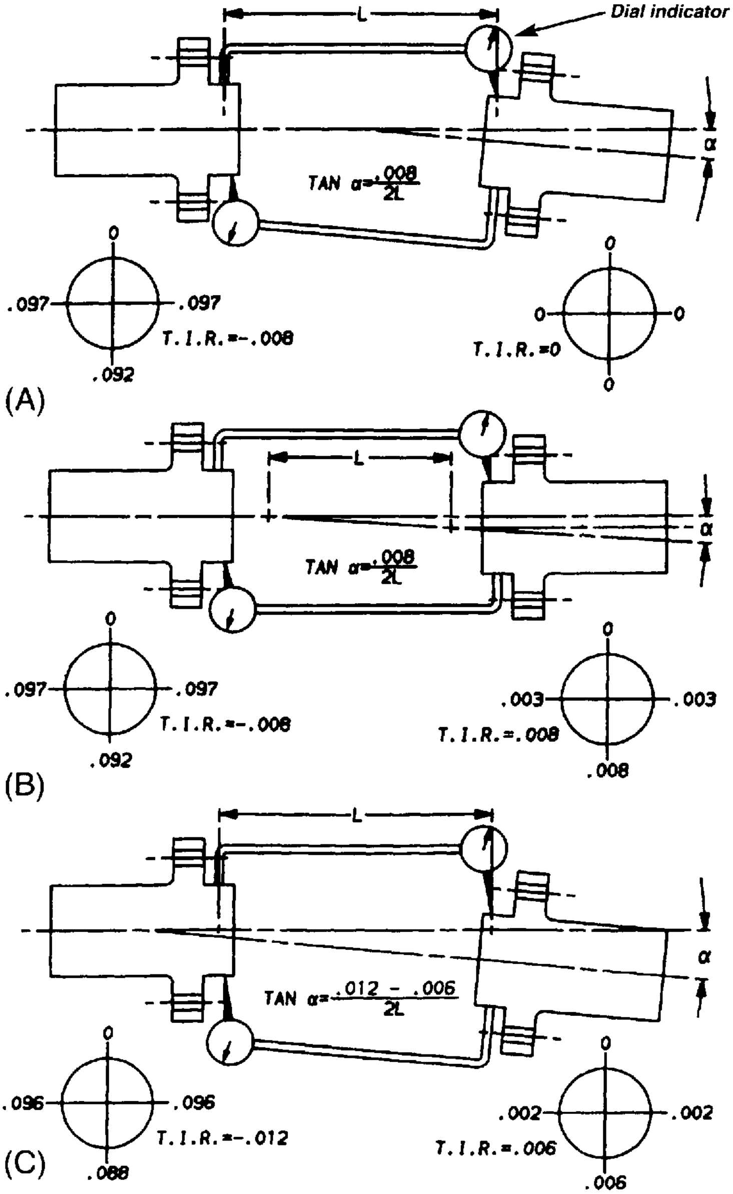


FIGURE 4.13 How to measure misalignment.

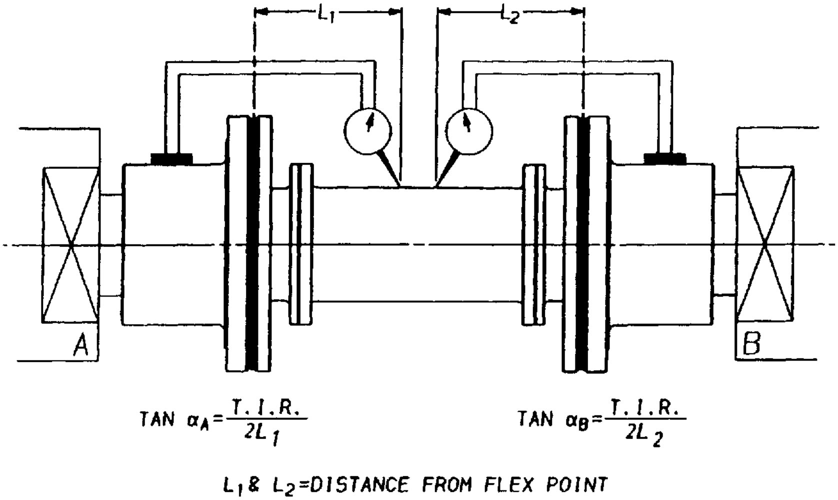

L_1
L_2
A
B
$\tan \alpha_A = \frac{T.I.R.}{2L_1}$
$\tan \alpha_B = \frac{T.I.R.}{2L_2}$
L_1 & L_2 = DISTANCE FROM FLEX POINT

Figure 4.14 Measuring misalignment on an assembled coupling.

H. How to Correct Misalignment

Attempting to correct the misalignment between two shafts through trial and error can be a lengthy and frustrating procedure. On the other hand, if a thorough procedure is followed, the job can be done with ease. The equipment required for measuring was described in the preceding section. The tools required for correcting misalignment are the same as those used to measure it, plus a pencil, paper, a good head for calculations, or a calculator, which would make it simpler and possibly be more accurate.

There are three important steps to follow when attempting to align equipment:

1. Rotate the shaft to which the indicator is attached, and lock or hold the other shaft stationary.
2. Correct the misalignment by moving the piece of equipment with the stationary shaft. Do not adjust the equipment to which the indicator is attached.
3. Make sure that the shafts have no axial play; lock or hold them in one position so that no false readings are taken.

1. Shaft Separation

The distance between the shafts (or flanges) is usually determined by the system designer but may be dictated by the type of coupling used. The required shaft separation is usually given on the coupling assembly drawing and with other instructions, which are usually packaged with the coupling.

Axial alignment of the equipment shafts should be the first part of an alignment procedure because it usually involves the greatest equipment movement. This begins with a rough alignment. If the shafts are close together, use a straightedge. If they are some distance apart, attach a dial indicator to the shaft that can easily be rotated, then move the other piece of equipment until the shafts are roughly in line. Following this, try to reduce the horizontal misalignment. Bring the indicator tip into a horizontal plane and set the dial at zero. Rotate the shaft 180° (one-half a revolution) and take a reading. If the reading is positive (an increase in the reading), move the equipment away from the indicator (move the stationary equipment); if the reading is negative, move the equipment toward the indicator.

EXAMPLE 4

The reading went from zero to 0.036 (clockwise). Therefore, we move the equipment away from the indicator by $0.036/2 = 0.018$.

EXAMPLE 5

The reading went from zero to 0.092 (counterclockwise). Therefore, we move the equipment toward the indicator by $(0.100 - 0.092)/2 = 0.004$.

2. Angular Misalignment

a. Face Readings. If the space between shafts allows the indicator to read on the face of the coupling hub, the angular misalignment can easily be corrected. However, there is always the possibility that the face of the hub is not square to the shaft. If that is the case, then even when the shafts are perfectly aligned you would obtain a false reading. To eliminate this "false" reading, rotate the shaft. If there is some wobble, record the value and use chalk or some other marker to mark the position of the maximum reading. Turn the shaft until the mark is on top. Now rotate the shaft that holds the indicator and zero the indicator when it lines up with the chalk mark.

EXAMPLE 6

The diameter D on which the tip of the indicator is rotated is 6 in., and the distance Y between the legs of the equipment is 15 in. (Figure 4.15). If the face runout is 0.006 and the TIR is 0.040 when the indicator shaft is rotated, then shim under the rear legs of the equipment by

$$\frac{0.040-0.006}{2}\times\frac{15}{3}=0.085\text{ in.}$$

If the indicator moved from zero to 0.060 (counterclockwise), the TIR is still 0.040. In this case, shim under the front legs of the equipment by

$$\frac{0.040+0.006}{2}\times\frac{15}{3}=0.115\text{ in.}$$

b. Horizontal Direction. If the distance between the shafts is too small for taking face readings, the angular misalignment can be measured and corrected through two successive readings in two locations w in. apart, starting with the indicator in a horizonal position. (See Figure 4.16.).

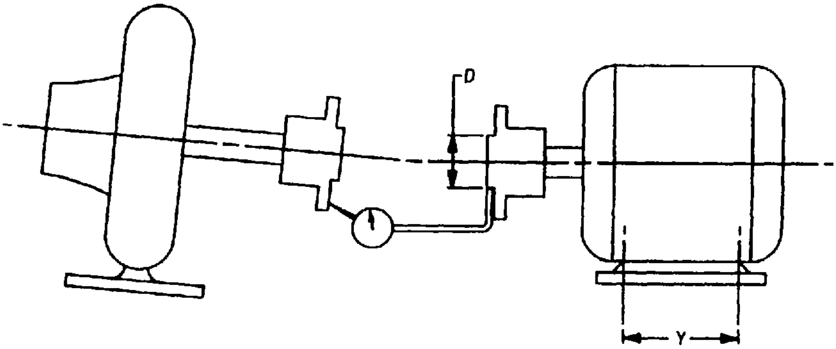


FIGURE 4.15 Correction of face runout.

EXAMPLE 7

In the first location the indicator moves from zero to 0.010 in. (clockwise); in the second location the indicator moves form zero to 0.016 in. (clockwise). If the distance $W = 4$ in. and $Y = 24$ in., move the rear end of the machine

$$\frac{0.016 + 0.010}{2} \times \frac{24}{4} = 0.018 \text{ in.}$$

away from the last position reading on indicator 2.

EXAMPLE 8

In the first location the indicator moves from zero to 0.082 in. (counterclockwise); in the second location the indicator moves from zero to 0.092 in. (counterclockwise). Move the rear end of the machine

$$\frac{0.092 + 0.082}{2} \times \frac{24}{4} = 0.030 \text{ in.}$$

toward the last position reading on indicator 1.

Note that when both readings are in the same direction, the small number is subtracted from the large one; when the readings are in opposite directions, they should be added.

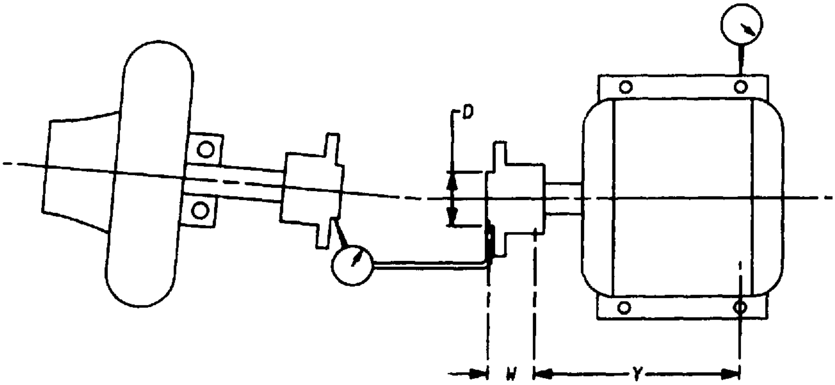


FIGURE 4.16 Correction for horizontal misalignment.

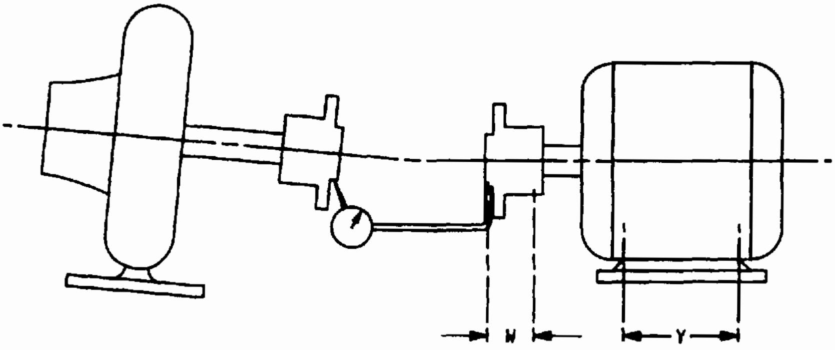


FIGURE 4.17 Correction of angular misalignment.

c. Vertical Direction. Basically, the same method is used as that described in the preceding section, except that the dimension Y is the distance between the machine legs (Figure 4.17). The indicator must always be zeroed when it is on top of the shaft.

EXAMPLE 9

In the first location the indicator moves to 0.006 in.; in the second location it moves to 0.021 in. If the distance $W = 4$ in. and $Y = 24$ in., then a shim should be placed under the front legs of the equipment to raise them by

$$\frac{0.021 - 0.006}{2} \times \frac{24}{4} = 0.045 \text{ in.}$$

EXAMPLE 10

In the first location the indicator moves to 0.087 in.; in the second location it moves to 0.098 in. A shim is required under the rear legs of the equipment to raise them by

$$\frac{0.098 - 0.087}{2} \times \frac{24}{4} = 0.033 \text{ in.}$$

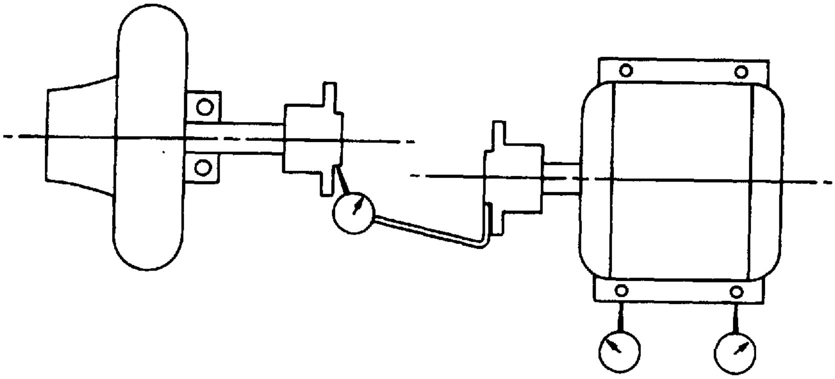


Figure 4.18 Correction for offset misalignment.

3. Parallel Misalignment (Offset)

For parallel misalignment (Figure 4.18), first repeat the rough alignment (see Section III. H. 1). To avoid inducing any angular misalignment, the machine should be moved parallel to itself. For this, install two indicators at the machine legs and move the machine in such a manner that both indicators always read the same.

Second, zero the indicators on top of the shaft and read after half a revolution (180°). If the indicator reads 0.062 in. (clockwise), then shim all four legs of the other machine by 0.062/2 = 0.031 in. If the indicator reads 0.068 in. (counterclockwise), remove

$$\frac{0.100 - 0.068}{2} = 0.016 \text{ in.}$$

from under each leg of the other machine. If shims cannot be removed, then add 0.016 in. shims under the legs of the machine on which the indicator is mounted.

4. Final Check

As you have probably noticed, until now all the indicator readings were taken with the indicator attached to only one shaft. Remember that this can lead to false alignment and that the reverse indicator method is best.

Now it is time to verify the accuracy of the alignment and perform a reverse indicator reading (Figure 4.13C).

If both readings show TIR = 0.018 in. (but one reading is clockwise and the other is counterclockwise), we have a pure parallel misalignment of 0.009 in. between the shafts. Compare your reading with the value allowed by the manufacturer of the coupling or equipment. Most coupling manufacturers give two misalignment capacities: one that is the maximum at which the coupling can operate and one that is acceptable for initial alignment. The latter is usually one-eighth to one-third of the operating capacity if no guide is given; then a value of less than one-eighth of operating capacity is usually acceptable. *Note:* The alignment tolerance acceptable to the equipment is normally smaller (see Figure 4.7).

I. Laser Alignment Systems

There are various types of laser alignment systems. Figure 4.19 shows the three basic measurement techniques used by all of them: the laser/detector/roof prism system, the dual laser/dual detector system, and the laser/beam splitter/dual detector system.

1. Advantages of Laser Measurement Systems

Bracket sag does not occur with laser beams, but the user must ensure that the brackets that hold the lasers and detectors are firmly attached to the shafts.

Accuracies of ±3 μm are attained.

Some systems allow for the coupling to be disconnected when capturing readings.

Most systems include an operator keypad/display module that prompts the user through the measurement steps and calculates the moves for one of the machines.

(Some systems can solve for a variety of movement solutions.)

2. Disadvantages of Laser Measurement Systems

Laser measurement systems are relatively expensive.

Their range of measurement is somewhat limited, since most manufacturers use 10 mm × 10 mm detectors.

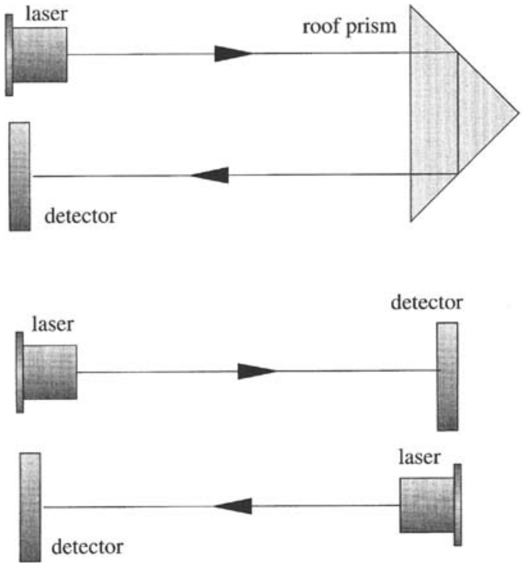


FIGURE 4.19 Basic principles of laser dector shaft alignment measurement systems.

They are incapable of measuring runout conditions (it is possible to align bent shafts and not know it).

Many systems determine moves for one of the two machines.

Most systems require that the coupling be bolted in place when capturing readings.

Both shafts have to be rotated.

It is sometimes difficult to capture readings in bright sunlight or well-lit areas.

Accuracy is reduced in the presence of excessive steam or heat.

Figures 4.20–4.24 show how alignment is done with a laser.

J. Hot Alignment

Machines are usually aligned when cold. Under normal operating conditions, some machines can become hot, and the alignment can change significantly due to thermal expansion of shafts and/or support struc-

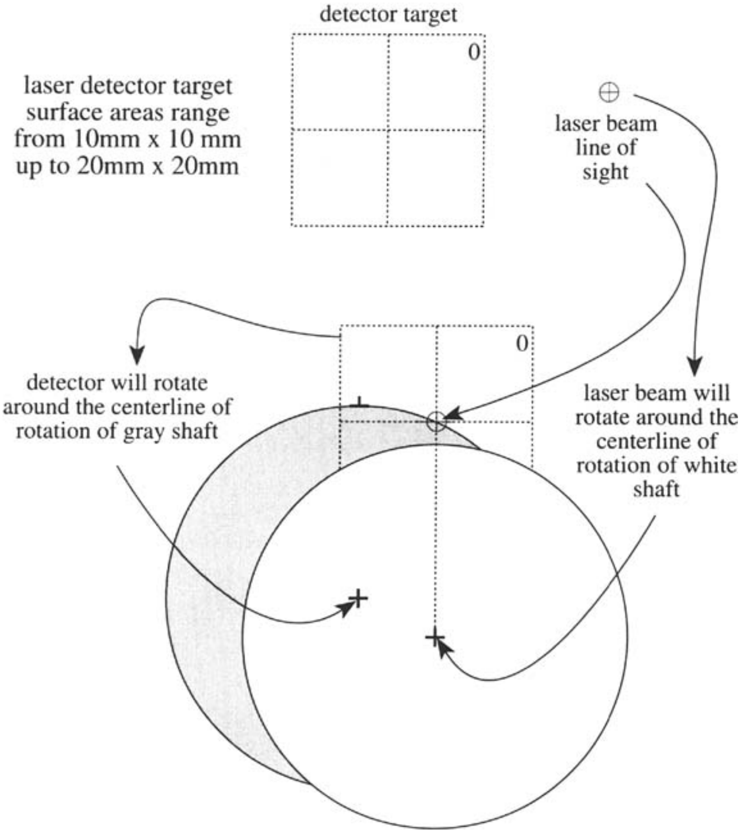

detector target
0
laser detector target surface areas range from 10mm x 10 mm up to 20mm x 20mm
laser beam line of sight
0
detector will rotate around the centerline of rotation of gray shaft
laser beam will rotate around the centerline of rotation of white shaft

FIGURE 4.20 What a laser detector "sees" looking down the axis of the shaft.

tures. This thermal movement must be measured by operating the equipment, then shutting it down and checking the alignment with the coupling installed. Alternatively, the information can be obtained from the machinery manufacturer or an alignment monitoring coupling can be used.

In any case, if the thermal movement is significant, zero cold alignment may not be beneficial. When thermal movements are large and have been accurately calculated or measured, the initial parameters should be such that alignment is as close to zero as possible when the

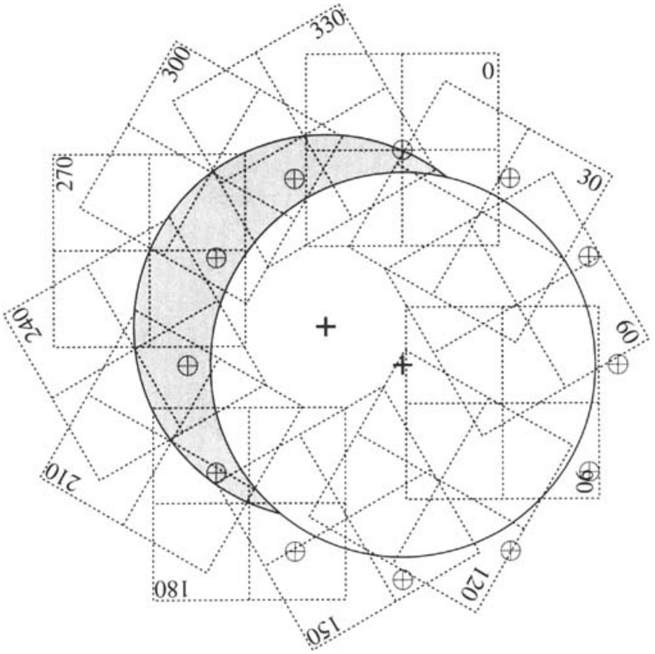
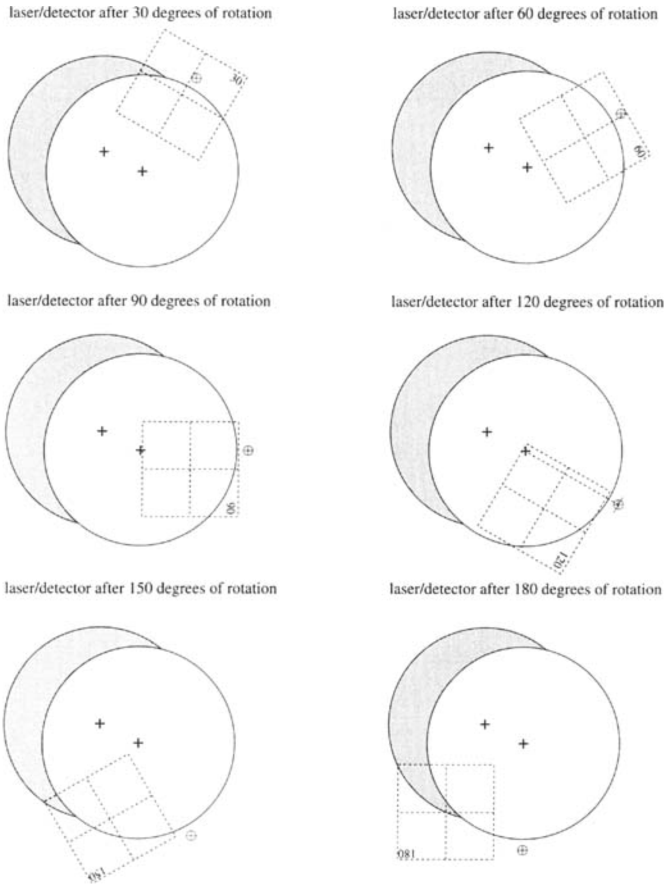


FIGURE 4.21 Where the laser beam strikes the detector through 360° of rotation.

equipment is in operation rather than when it is cold. This means that during cold alignment the coupling is usually purposely misaligned to compensate for thermal effects during operation.

K. Vibration Signatures

Vibration can and does occur on rotating machinery that is being subjected to misalignment conditions. Data shown below were taken from *Shaft Alignment Handbook*, by John Piotrowski (Second Edition, Marcel Dekker, 1995). Over a ten year period, several individuals have conducted controlled tests on rotating machinery where the drivetrain was subjected to misalignment conditions and vibration data were collected and observed. In addition, data were collected on machinery where there was a known misalignment condition present. Figures 4.25–4.29 show samples of these data for:



laser/detector after 30 degrees of rotation

laser/detector after 60 degrees of rotation

laser/detector after 90 degrees of rotation

laser/detector after 120 degrees of rotation

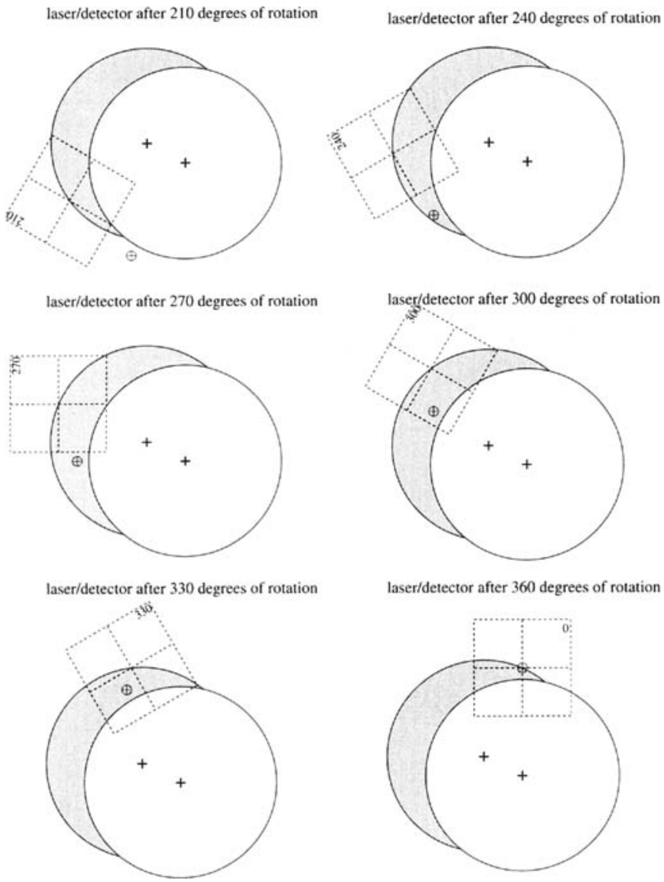
laser/detector after 150 degrees of rotation

laser/detector after 180 degrees of rotation

FIGURE 4.22 Where the beam strikes the detector through 30–180° of rotation.

1. Flexible disc coupling (see Figure 4.25, p. 217)
2. Gear coupling (see Figure 4.26, p. 218)
3. Grid coupling (see Figure 4.27, p. 219)
4. Jaw coupling (see Figure 4.28, p. 220)
5. Rubber tire coupling (see Figure 4.29, p. 221)

As you review these figures you will notice that the vibration patterns



laser/detector after 210 degrees of rotation

laser/detector after 240 degrees of rotation

laser/detector after 270 degrees of rotation

laser/detector after 300 degrees of rotation

laser/detector after 330 degrees of rotation

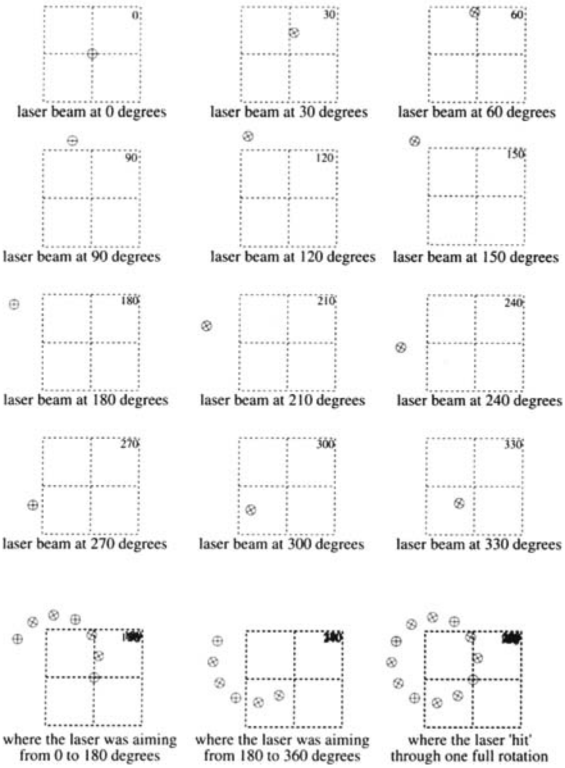
laser/detector after 360 degrees of rotation

Figure 4.23 Where the laser beam strikes the detector through 210–360° of rotation.

on equipment with the same coupling design are not exactly the same and in some cases are radically different or nonexistent.

IV. COUPLING ASSEMBLY

Once the hubs are installed on the shafts and the equipment is aligned, the coupling can be assembled. Each type of coupling is assembled differently; however, the general procedure is the same for many of the



laser beam at 0 degrees

laser beam at 30 degrees

laser beam at 60 degrees

laser beam at 90 degrees

laser beam at 120 degrees

laser beam at 150 degrees

laser beam at 180 degrees

laser beam at 210 degrees

laser beam at 240 degrees

laser beam at 270 degrees

laser beam at 300 degrees

laser beam at 330 degrees

where the laser was aiming from 0 to 180 degrees

where the laser was aiming from 180 to 360 degrees

where the laser 'hit' through one full rotation

FIGURE 4.24 What we would see if we were standing next to the detector as the shaft was rotated.

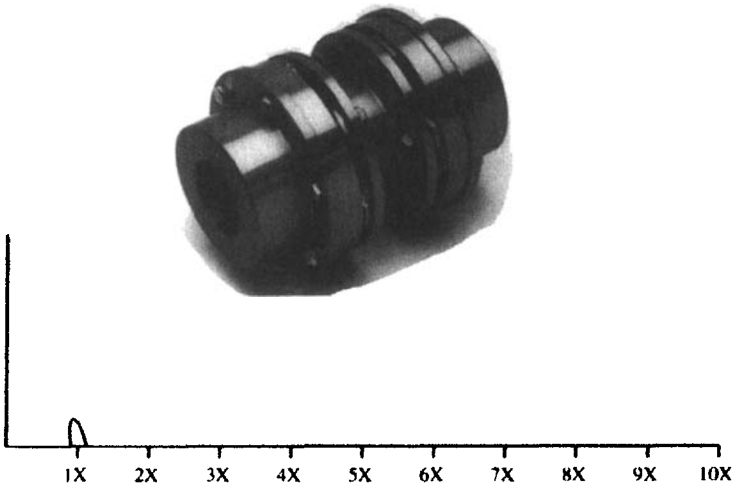
couplings. The following discussion covers some general procedures for several of the most common types of couplings. For more specific instructions and guidelines, consult the specific coupling manufacturer's instructions, which are usually shipped with the coupling.

A. Lubricated Couplings

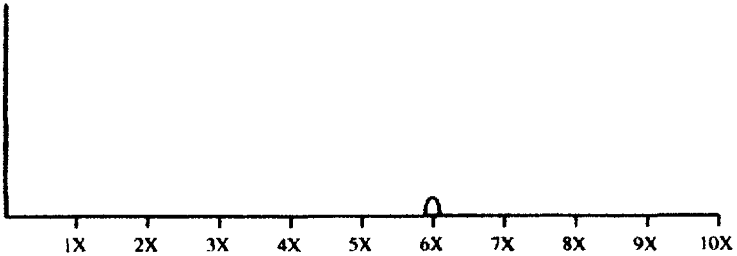
There are four types of lubricated couplings, which are illustrated in Chapter 2:

1. Gear type (Figures 2.16 and 2.17)
2. Chain type (Figure 2.18)

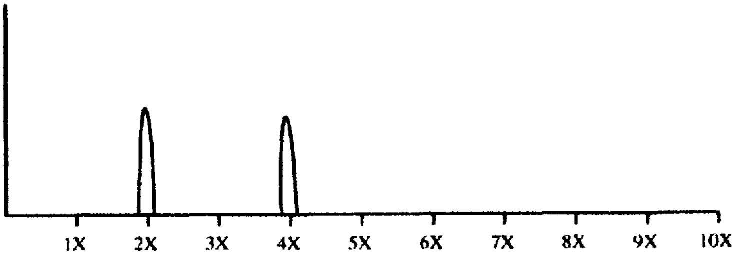
(*continues on p. 220*)



Motor driven BFW pump
Motor driven demonstrator
J. Piotrowski horizontal misalignment 80 mils IB & OB

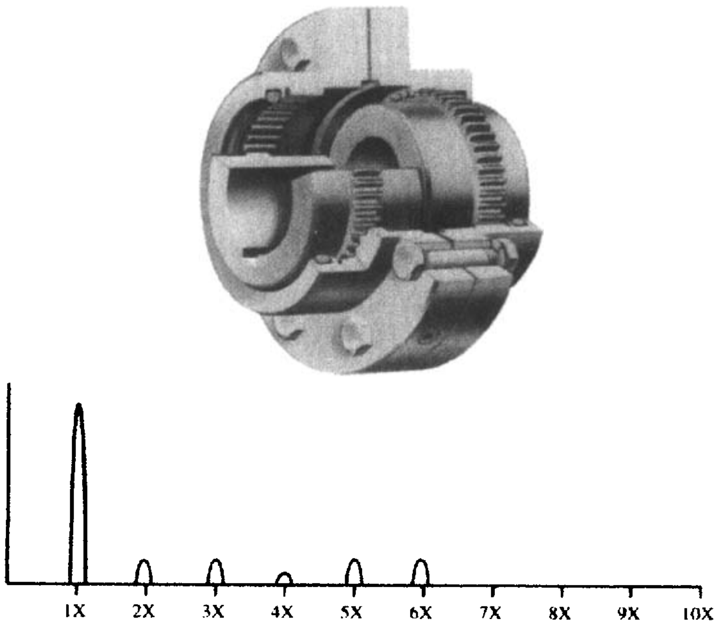


Motor driven generator test
D. Nower horizontal & angular misalignment 75 mils high

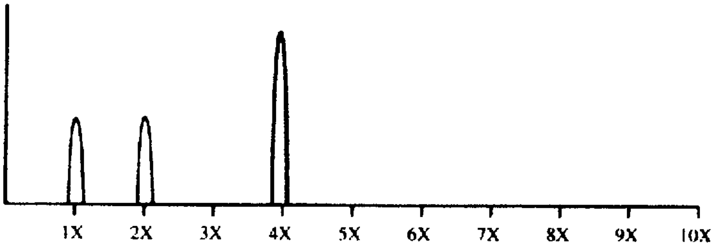


Motor driven motor experimental test
D. Dewell parallel @ 96 mils

FIGURE 4.25 Observed vibration patterns on flexible disc-type couplings.

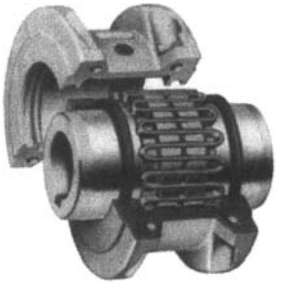


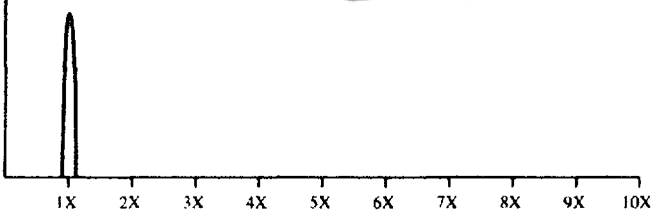
Motor driven ANSI pump
J. Lorenc horizontal misalignment 30 mils IB & OB



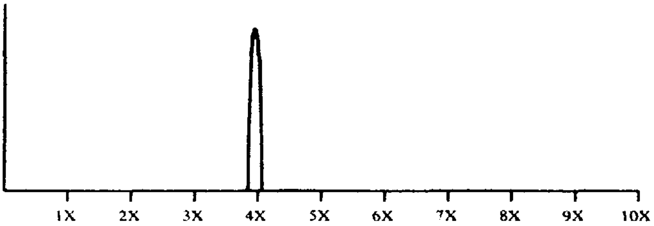
Gas/Power Turbine driven compressor
J. Piotrowski horizontal misalignment 65 mils IB & OB

Figure 4.26 Observed vibration patterns on gear-type couplings.

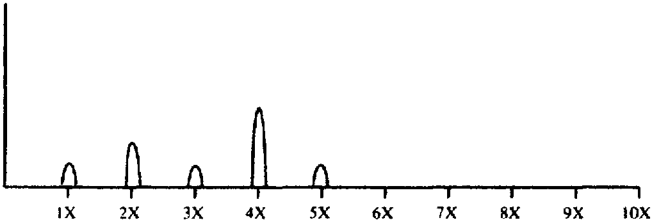




Motor driven ANSI pump
S. Chancey vertical misalignment 50 mils @ IB & 75 mils @ OB
J. Lorenc horizontal misalignment 90 mils IB & OB

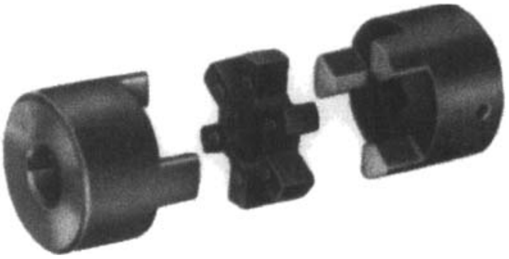


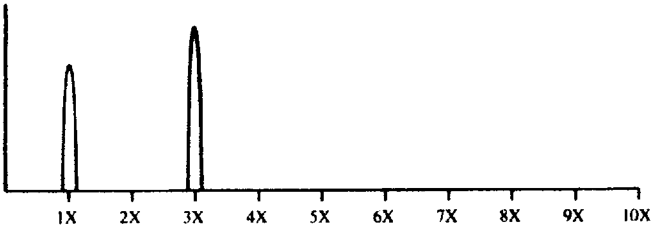
Motor driven generator test
D. Nower horizontal misalignment 50 mils IB & OB



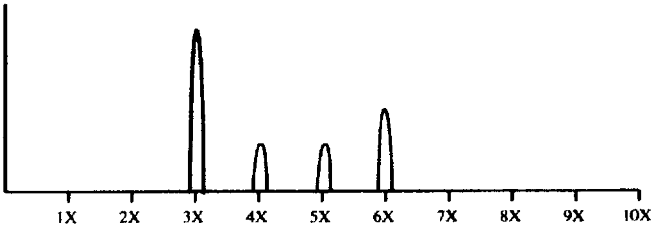
Motor driven centrifugal pump
J. Piotrowski horizontal misalignment 36 mils IB & OB

FIGURE 4.27 Observed vibration patterns on grid-type couplings.





Motor driven ANSI pump
J. Lorenc horizontal misalignment 90 mils IB & OB



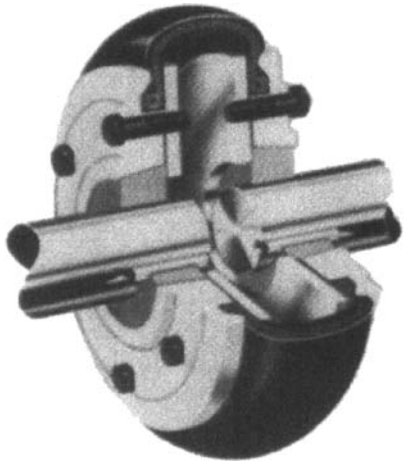
Motor driven generator test
D. Nower horizontal & angular misalignment @ 15 mils per inch

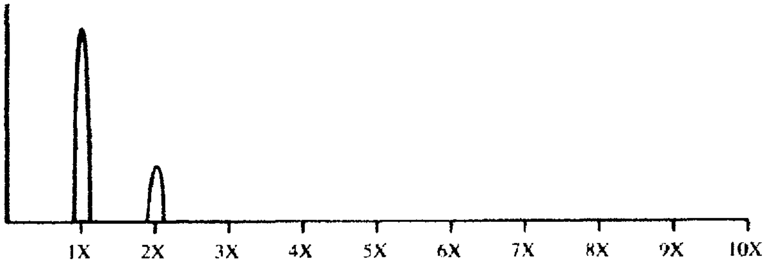
Figure 4.28 Observed vibration patterns on jaw-type couplings.

3. Steel grid type (Figures 2.20 and 2.21)
4. Universal joint (Figures 2.24, 2.44, and 2.45)

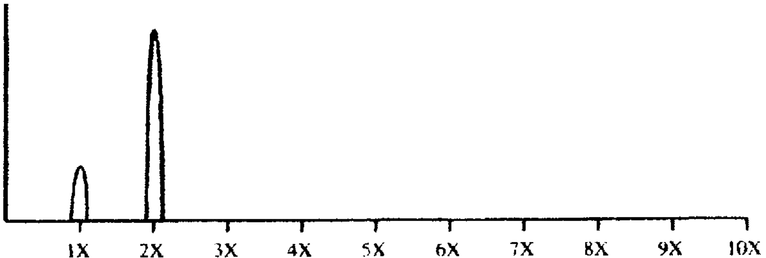
The gear, chain, and steel grid are very similar and are covered together.

The first step is the application of a coating of lubricant to all moving parts, such as gear teeth and chains or steel grids as well as to the O-rings and their mating surfaces. Then the coupling can be closed by





Motor driven ANSI pump
J. Lorenc horizontal misalignment 90 mils IB & OB
J. Piotrowski horizontal misalignment 80 mils IB & OB



Motor driven generator test
D. Nower horizontal & angular misalignment 75 mils high

FIGURE 4.29 Observed vibration patterns on rubber couplings.

bringing the sleeves or the covers together. Most lubricated couplings use gaskets or O-rings between the flanges so that the lubricant is sealed in. The importance of properly sealing a coupling cannot be overemphasized, because the amount of lubricant contained in a coupling is small. Even the smallest leak would soon deplete the coupling of lubricant, resulting in rapid wear and ultimate failure.

To ensure proper sealing, the flange surfaces must be clean, flat, and free of burrs or nicks. Any imperfections on the surfaces of the flange must be removed by using a honing stone or fine-grit sandpaper.

The gaskets are made of special material, as oil can seep though regular cardboard or paper. The gasket should be in one piece and be free of tears or folds. To facilitate assembly, some people glue the gasket to one of the flanges with grease. This procedure should be avoided, as it can prevent proper tightening of the bolts. It is suggested that a thin oil be applied to the gasket, which will help hold it in place.

Once the flanges are brought together, insert the bolts in the holes. The bolts must be assembled from the flange side, as shown on the assembly drawing. The holes in the flanges must be oversized and also countersunk to accept the bolt-head radii. Take care to align the holes in the gaskets with those in the flanges. If this is not done, the gasket will be torn by the bolts and pieces of material can stick between the flanges, preventing proper sealing.

Once all the bolts are in place, slide on the lockwashers and install the nuts. If self-locking nuts are used, lockwashers are not used. If room permits, the nuts (not the bolts) should be tightened to specifications, which means that a torque wrench should be used. The nuts should be tightened in two or more steps in a crisscross fashion, similar to the way automobile lugs are tightened.

If the sleeves or covers incorporate lube plugs, the plugs on the two sleeves should be diametrically opposite. If the coupling was dynamically balanced, match marks may indicate the relative position not only between the two sleeves (or covers) but also between the sleeves and the hubs, unless the coupling was component-balanced, in which case alignment may not be required.

Care should be exercised not to mix parts of similar couplings unless the coupling manufacturer's instructions say that it is acceptable to do so. Pay particular attention to balanced couplings and special couplings, where parts mixing may affect not only balance but also the functional operating characteristics of the coupling.

Universal joints are simple to install. They are usually shipped with their center section assembled and are ready to be placed between mounted hubs or flanges on the equipment. Caution should be exercised in slinging and handling universal joints. The joints should be slung from yokes; to prevent damage to sliding parts, do not sling around the shaft or splined protector. Joints should always be transported and handled in the horizontal position; *do not hang or transport joints in the vertical position.* Prior to installation, carefully clean the universal joint, flanges, and mating flanges, Remove all traces of rust proofing. Check the key fit in the shaft's rigid hubs. Install the rigid hubs on the shaft; for shrink fits, apply heat to the hubs uniformly. Check the mating flange for face runout and pilot runout. Check the pilot and faces for nicks and dents and stone any that you find. Check the universal shaft to ensure that match marks line up. Check the shaft-to-shaft or flange-to-flange dimensions between pieces of equipment to ensure that they conform to the dimensions supplied on the drawing. Bolt the mating flanges. Lubricate the bearing and/or length compensator unless this was done at the factory. Refer to specific instructions to verify this.

B. Metallic Flexible Element Couplings

1. Flexible Disc Couplings

Flexible disc couplings come in two basic configurations.

Low-speed discs (Figure 2.32) are usually taped or wired together. The discs are alternately fastened to the hub and spacer by bolts, nuts, and sometimes standoff spacers. If standoff washers are used, they must be assembled with the correct surface against the disc (usually curved). Sometimes washers of different thickness are used. The washers must be placed in the correct position. See the specific instructions.

High-speed discs (Figure 8.33) are usually assembled as a pack, and the pack is bolted alternately to the hub and spacer.

2. Diaphragm Couplings

Diaphragm couplings come in two basic configurations.

A flexible spacer section coupling (Figure 2.53) just drops in between the two installed rigid hubs. The center section is one-piece: a subassembly or a welded assembly. The center section should not be disassembled unless so specified by the coupling manufacturer's instructions. Some couplings have a pilot rigid hub, which requires that the center section be compressed for assembly.

On a flexible subassembly diaphragm coupling (Figure 2.54) the center of the diaphragm coupling section consists of three separate pieces or assemblies. This construction is common where equipment is separated by a long distance, as it helps to reduce the cost of spares.

C. Elastomeric Couplings

The rubber donut coupling (Figure 2.28) is assembled with radial bolts, and in the process of tightening the bolts the rubber "legs" are precompressed. Precompression should be uniform around the coupling. The split insert should be the first to reach the final "seated" position in the hub's notch.

Rubber tire couplings (Figures 2.25 and 2.26) are similar to donut couplings. They have a split element that is wrapped around the hubs. However, the fasteners are arranged axially, and they clamp the side walls of the tire between two plates.

Pin-and-bushing couplings, as well as jaw-type couplings (Figures 2.30 and 2.31), have an elastomeric element trapped between the hubs. To install or remove the element, usually at least one of the hubs slides on the shaft; if not, the equipment must slide axially out of the way. The shaft separation must be greater than the thickness of the element to allow for installation; however, some elements are split so that close shafts can be used.

Most elastomer-type couplings can be obtained in a spacer configuration (Figure 7.28), in which the two hubs and the flexing element can be "dropped out" as a unit. This unit is attached to the two shafts through two "rigid" hubs with axial bolts. To install tile couplings between the rigid hubs, the element has to be squeezed so that it will fit between the rabbets of the rigid hubs.

D. Bolt Tightening (Flange Connections)

Most instruction sheets provided with couplings give information on tightening bolts for flange connections (see Figure 4.30). Unfortunately,

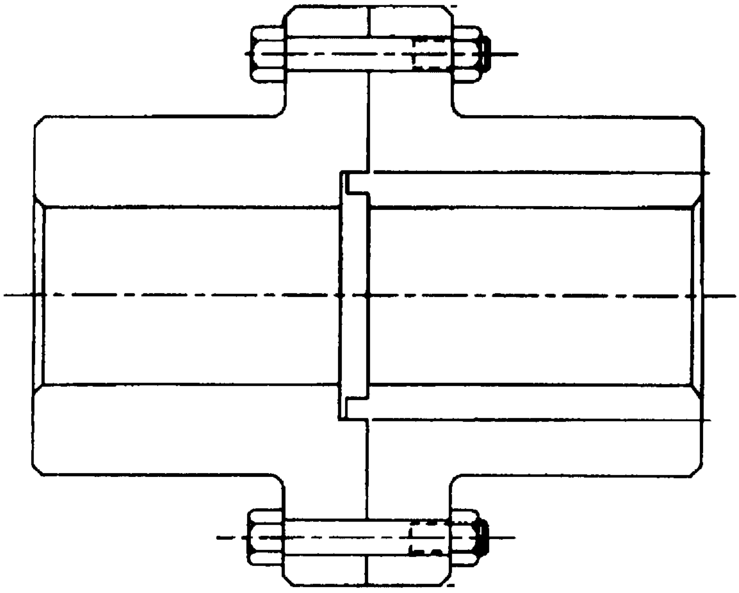


FIGURE 4.30 Bolted flanges.

these instructions are not always followed; many mechanics tighten the bolts by feel.

There are many important reasons why bolts should be properly tightened. Here are a few:

Couplings resist misalignment, and the resulting forces and moment put a strain on the equipment and connecting fasteners. If the fasteners are loose, they are subjected to alternating forces and may fail through fatigue.

A bolt that is not properly tightened can become loose after a short period of coupling operation.

If the pretorque is less than recommended, the bolt stretch is not correct and the bolt may not stay tight.

Few bolts work only in tension. Most coupling bolts also work in shear, which is caused by the torque transmission. Usually, only some of the torque is transmitted through bolt shear; part of the load is transmitted through the friction between the flanges.

Depending on the coupling design, as much as 100% of the torque can be transmitted through friction. If the bolts are not tightened properly, there is less clamping force, less friction, and more of the torque is transmitted through shear.

Because of the combined shear and tensile stresses in bolts, recommendations for bolt tightening vary from coupling to coupling. Coupling manufacturers usually calculate bolt stresses, and their tightening recommendations should always be followed. If recommendations are not available, it is strongly suggested that a value be obtained from the coupling manufacturer rather than guessing. *Find out what the specific coupling requires.*

Bolts should be tightened to the recommended specification in at least three steps. First, tighten all bolts to one-half to three-fourths of the final value in a crisscross fashion. Next, tighten them to specifications. Finally, recheck the first bolt tightened to the final value after all the bolts have been tightened. If more tightening is required, recheck all the bolts. Also, the higher the strength of the bolt, the greater the number of steps you should take in tightening all bolts:

Grade 2:	Two or three steps
Grade 5:	Three or four steps
Grade 8:	Four to six or more steps

If an original bolt is lost, *do not* substitute a commercial bolt just because it looks similar to the other coupling bolts. It is best to call the coupling manufacturer to order another bolt or ask if they can suggest an alternative.

If room permits, always tighten the nut, not the bolt. This is because part of the tightening torque is needed to overcome friction. The longer the bolt, the more important it is to tighten the nut rather than the bolt. As there is additional friction when turning the bolt, more of the effort goes into friction than into stretching the bolt.

Some couplings use lockwashers; others use locknuts. Whereas a nut–lockwasher combination can usually be used many times, a locknut loses some of its locking properties every time it is removed from the bolt. If not instructed otherwise, it is best to replace hex locknuts after five or six installations. Some couplings use aircraft type bolts with either hex locknuts or 12-point nuts. In this case the hex locknuts should be replaced after five or six times and the 12-point locknuts should be replaced after 10–15 times (usually when they lose their locking features).

V. LUBRICATION OF COUPLINGS

Since a large number of flexible couplings require lubrication, a separate section is being devoted to lubrication. All couplings classified as mechanical require lubrication. These couplings are

1. Gear couplings
2. Chain couplings
3. Grid couplings
4. Universal joints

These four types of lubricated flexible couplings, although different, have very similar mechanisms of lubrication. The gear coupling is generally used as an example in this section; however, most discussions apply to chain and grid couplings also.

The gear coupling has four or five major parts: two hubs, two sleeves, and in many applications a spacer. It also has bolts and nuts and must have some type of seal to ensure that the lubricant stays in.

Each gear mesh acts like a spline. Some gear teeth, both internal and external, are straight, and some couplings have straight internal and external teeth with crowns. The mesh has clearance (backlash). Misalignment results from sliding or rolling these loosely fitted parts. Because of this relative motion, the need for lubrication is clear. In some flexible couplings, motion amounts to only thousandths of an inch, but in some it may be several inches (4–6 in.). From this range of motion a coupling can experience sliding velocities from less than 1 in./sec (ips) to as much as 200 ips.

It is important to remember the effect of centrifugal forces on lubricants. In a gear coupling the lubricant is trapped in the sleeve, which at high speeds acts as a perfect centrifuge. It tends to separate the grease, and for continuously lubricated couplings it will separate foreign particulates out of the lubricant, which tend to build up in the coupling to form sludge. Chain and grid couplings are much like gear couplings except that the "sleeve" in one case is the chain and cover and in the other case the grid and cover.

The universal joint acts slightly differently. If a telescoping spline is used, the spline acts like a gear coupling and must be lubricated like one. In the yoke heads there are bearings (needle or roller type) that are lubricated by the rolling action of these bearings. Since most universal joints operate at relatively low speeds, centrifugal force usually has very little effect on lubrication.

A. Types of Seals

For lubricated couplings the seals are almost as important as the lubricant itself. Almost as many coupling failures can be attributed to seal failure as to lube failure.

There are two basic types of seals used to retain lubricants: the metallic labyrinth seal and the elastomeric seal. Again we will use the gear coupling to discuss specifics, but most of these sealing devices are also incorporated in the chain and grid couplings and the universal joint.

1. Metallic Labyrinth Seals

When gear couplings were introduced in the 1920s, synthetic positive seals, which can withstand repeated flexing motion between the hub and the sleeve and the deterioration caused by the lubricant, were not available. Thus, as shown in Figure 4.31A, the all-metal labyrinth seal plate with its center of contact in line with the center of the gear flex point was used. Metallic seals usually require clearance and are not considered to be "positive" seals. Lubricant can leak out of a coupling if coupling motion is stopped and or is constantly being reversed. Also, this seal does not provide adequate protection in keeping couplings free of contaminants that could cause the coupling or its lubricant to deteriorate. This type of seal is still used very successfully in many applications.

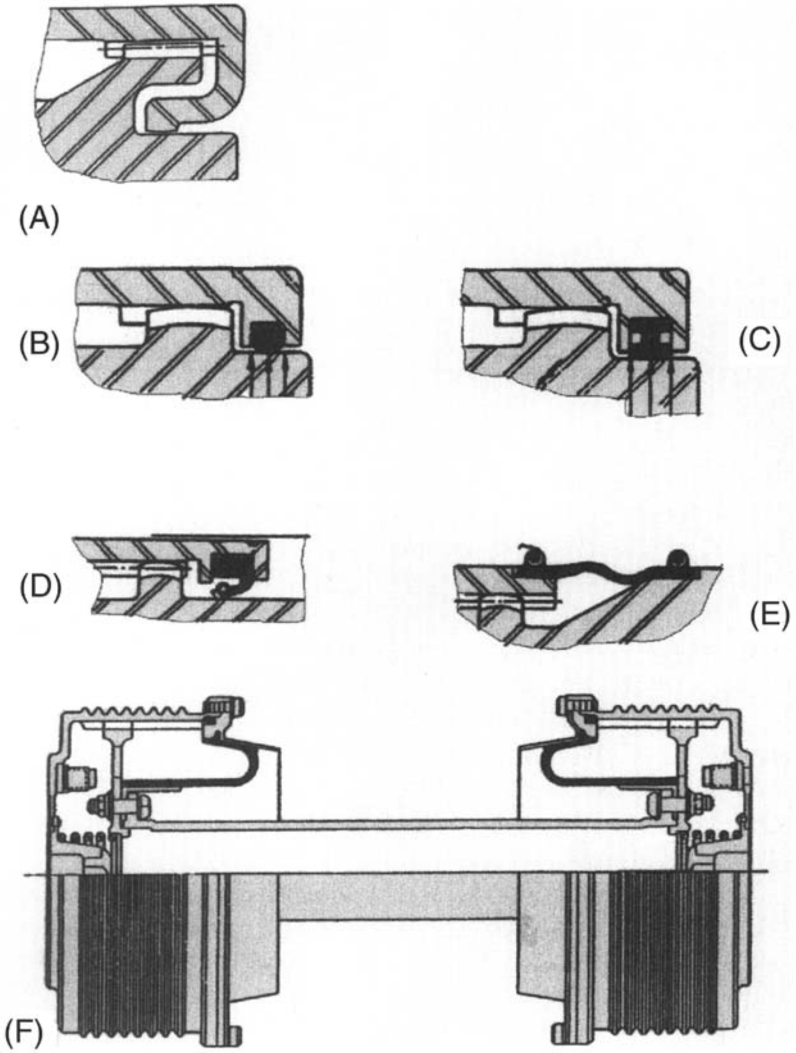


FIGURE 4.31 Types of seals, (A) Metallic labyrinth seal. (B) O-ring seal. (C) H or T cross-section seal. (D) Lip seal. (E) Boot seal. (F) Fabric-reinforced boot seal.

2. Elastomeric Seals

There are four basic elastomeric seals used in couplings:

1. The O-ring
2. The H or T cross-section seal
3. The lip seal
4. The boot seal

The O-ring seal shown in Figure 4.31B was developed by the U.S. Air Force during World War II and is now very common in mechanical equipment. Most couplings requiring lubrication use O-rings, which are one of the most positive means of retaining lubricant. The major disadvantage with O-rings is that they usually allow relatively small amounts of motion between parts. To account for the oscillatory motion, seals can be squeezed through only approximately 10% of their cross section before they start to take a set and no longer function.

The H or T cross-section seal (Figure 4.31C) functions and operates very similarly to an O-ring but has more surface area in contact with the sealing surface and for relatively small motion will usually do a slightly better job. In many applications moderate amounts of oscillatory motion occur together with axial movement, and the H or T cross section seal tends to roll and twist. When this occurs, a spiral path is created through which the lubrication can exit the coupling. Even the smallest leak would soon deplete the coupling of lubricant, resulting in rapid wear and ultimate failure.

The lip seal (see Figure 4.31D) is another very common seal used with couplings. This seal can be designed with "extended" lips that can accommodate high eccentricities and thus a large degree of misalignment. It can also be designed with spring retainers and steel inserts to prevent the lips from lifting off at high speed.

The boot seal (see Figure 4.31E) is not in very common use, but it is a very effective means of retaining lubricant. This type of seal is usually limited to operation at low speed, but with a special fabric reinforcement material (see Figure 4.31F) it has been used on couplings that operate at 7 1/2° of misalignment and 6000 rpm.

B. Methods and Practices of Lubrication

1. Grease Lubrication

There are two methods for lubricating flexible couplings with grease: lubrication before closing the coupling (pack lubrication) or lubrication after closing the coupling. To pack-lubricate a coupling, an appropriate quantity of grease is applied manually to each half-coupling, making sure that the teeth and slots are coated. Then the coupling is bolted up or assembled. If couplings are supplied with lube plugs, the working surfaces are wiped with a light coating of lubricant before the coupling is assembled. Usually, couplings are provided with two lube plugs in their periphery; both should be removed before lubrication is attempted. Only the amount of lubricant specified by the coupling manufacturer should be used, as too much could cause seals to be damaged or cause the coupling to bind up in operation. After filling, the lube plugs are replaced.

Periodic maintenance of grease-lubricated couplings is necessary if the coupling is to give satisfactory service. A coupling should be relubed at regular intervals, usually every 6–18 months. The coupling manufacturer's suggestions should always be followed. Most relubrication cycles are established by experience, so records should be kept of relubrication history.

It is also suggested that the couplings be disassembled and cleaned at regular intervals to get rid of foreign materials and to check for wear and deterioration. Parts should be washed, dried, and inspected. If parts appear to be worn, they should be replaced, relubricated, and reassembled.

Note: When disassembling and relubricating a coupling, inspect it to check the condition of the lubricant. If the grease is soapy or has separated into oil and soap, it is evident that this lubricant is not suitable for this application. It would be wise to contact the coupling manufacturer and obtain a better lubricant recommendation.

2. Oil Lubrication

a. Continuous Flow. In the continuous flow method of lubrication, oil is injected into the coupling by one or more oil jets directed toward the collecting lips (which can be either fixed or removable) or rings. Oil may be injected at one end of the coupling and exit at the other, or each end can be lubricated separately. Oil collectors may be provided on the hubs, sleeves, or spacer, depending on the coupling design. Oil jets are made from tubes that have been capped and then drilled to provide the correct quantity of oil and the correct exit velocity based on supply oil pressure and the size of the supply line. The oil flow for a given type of design is a function of coupling size, transmitted horsepower, speed, and possible coupling misalignment. Figure 4.32 shows a curve for a high-speed gear coupling that gives the lube flow requirements as a function of horsepower and misalignment. Such curves differ for dif-

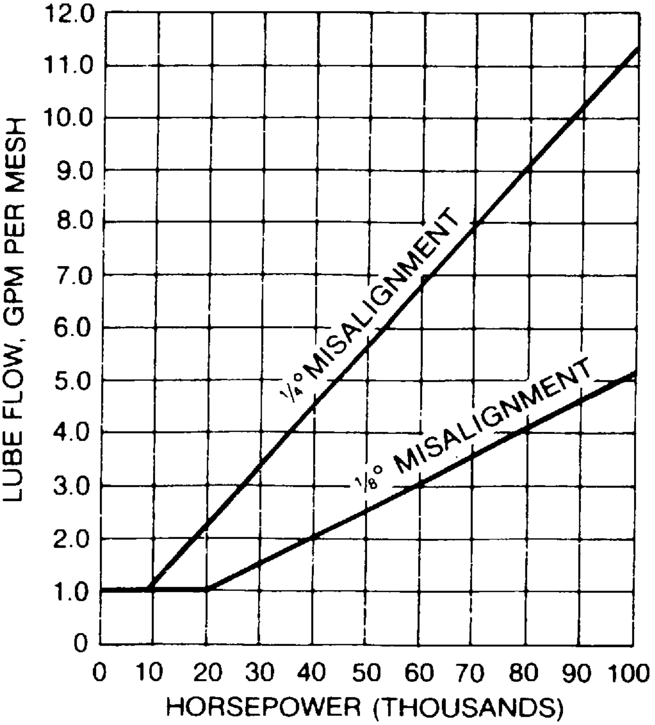


FIGURE 4.32 Amount of lube flow required for gear couplings as a function of horsepower and misalignment.

ferent types of couplings and also from one manufacturer to another. Use whatever is recommended by the specific coupling manufacturer.

Several things determine the lubricant flow required. Material hardness is one; another is whether the coupling incorporates positive dams or is damless. Damless couplings usually require much more oil. Usually, continuously lubricated couplings use the same oil as that used on the connected machinery bearings. It is mandatory that a sufficient quantity of clean oil (filtered, not contaminated with water or corrosive media) be supplied to the coupling. It is suggested that a separate filter be used, usually 5 μm in size or smaller, and that the oil properties and contamination be monitored and a centrifuge be used.

Sludge accumulation (Figure 4.33) may be detrimental to equipment and the coupling. It can produce high moments and forces on the equipment and the coupling that can result in catastrophic failure. Sludge may affect a coupling in several ways: It can reduce axial movement between coupling parts, it tends to accelerate wear, and it can stop

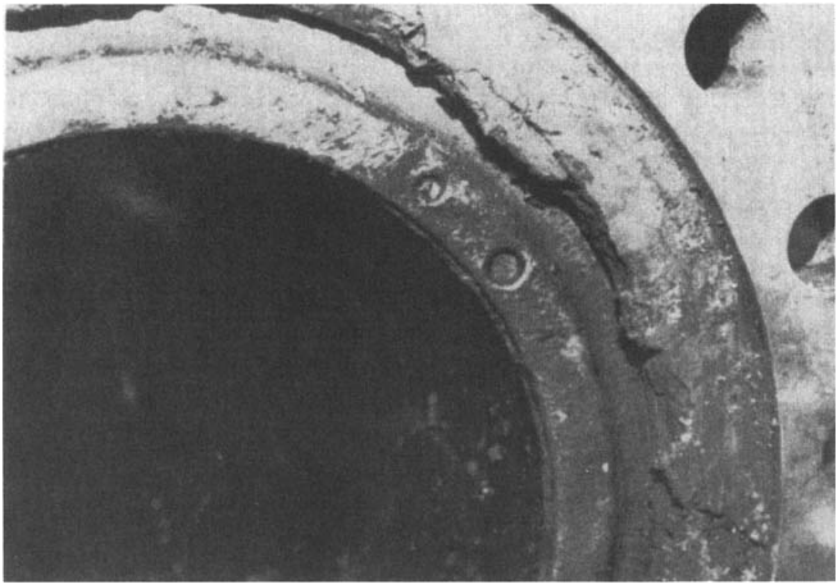


FIGURE 4.33 Sludge-contaminated gear coupling.

the flow of oil, causing the coupling to operate without sufficient lubricant and, in a contaminated system, cause or encourage corrosion. Sludge can be controlled by filtration to remove foreign particles, and lube contamination by water and corrosive media can be prevented by the use of separators and constant sampling of the lube and/or by sealing the coupling housing system from the contaminated atmosphere.

Most couplings are designed to minimize sludge (see Figure 4.34). Many people believe that the solution to sludge accumulation is to remove the coupling dams. Unless you can dramatically increase the coupling lubricant flow and ensure a reliable lube source, the problems associated with damless couplings usually far exceed the possible benefits. Figure 4.35 shows what happens to the oil in a damless coupling. Unless the lubricant flow is substantially increased, most of the working portion of the teeth will be starved. If couplings are properly maintained, the sludging problem can be minimized. It is suggested that the coupling be opened and inspected for sludge accumulation at regular intervals and every time the equipment is stopped. The sludge should be removed mechanically and/or chemically, and the parts should be thoroughly cleaned and inspected for damage (tooth distress, corrosion, breakage, etc.).

b. Confined and Self-Contained. Whenever a coupling is confined in a housing together with other lubricated components such as gears, it

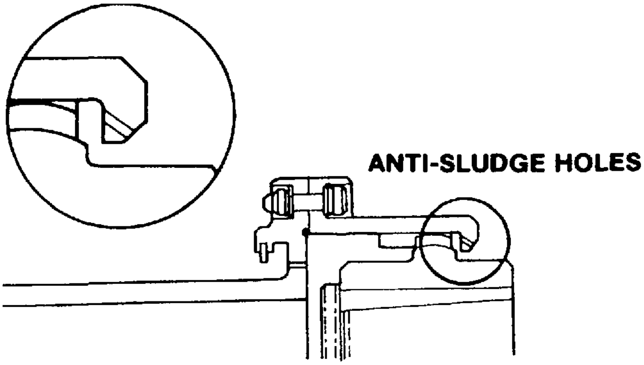


FIGURE 4.34 Design to help minimize sludge buildup—flow-through design.

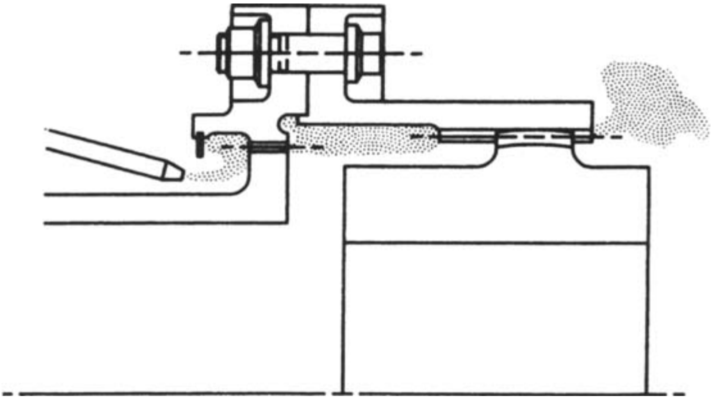


FIGURE 4.35 Oil flow in a damless gear coupling.

may use the oil from the housing for its lubrication. When the coupling operates above the oil surfaces, lubrication is provided either by an oil pump or through splashing. In either case the oil flow to the confined coupling is a function of operating speed. If the working surfaces of a coupling are partially or totally submerged in oil, the lubrication requirements are satisfied by the flow of oil from the housing into the coupling. To ensure adequate lubrication, submerged couplings are provided with holes in their covers or sleeves, or the coupling may incorporate pickup scoops or other arrangements to aid in ensuring adequate lubrication. Because of their centrifugal action, these types of couplings tend to retain dirt from the system; it is therefore advisable to clean the coupling when the equipment is down or during scheduled periodic cleaning intervals.

Self-contained couplings are supplied with seals and can be filled at assembly and operated for an extended period without further attention. The usual method of filling this type of coupling is to remove the lube plugs and pour the correct amount of recommended oil into the coupling. It is recommended that the working portions of the coupling be coated with lubricant before assembly. This will prevent bare metal contact between mating parts at assembly and at startup before the oil is properly distributed. Maintenance of an oil-filled coupling consists basically of preventing the oil film from being lost from the coupling. It is therefore important to maintain flange surfaces, seals, and gaskets in good condition. Lube plugs should be properly seated (or sealed) so that no leakage occurs. A regular lubrication schedule should be established and maintained, following all of the manufacturer's assembly, lubrication, maintenance, and cleaning instructions.

C. Types of Lubrication

Couplings are usually lubricated with oil or grease. Lubricated couplings are usually either self-contained or lubricated from an external supply. For a self-contained coupling, oil or grease can be used and covers and/or seals are required. For an externally lubricated coupling, the lubricant is oil. The oil is supplied at a specific flow rate, or the coupling may be dipped or intermittently lubricated.

Which type of lubrication is best, sealed lube or continuous? Gear, chain, and grid couplings can be filled with a specific quantity of lubricant and sealed. Many factors govern which method should be chosen, and each coupling application must be evaluated with respect to its particular requirements. The following are some of the factors that should be considered.

Sealed Lubrication

1. Sealed lubrication affords the opportunity to choose the best lubricant available.
2. Outside contaminants are effectively prevented from entering working surfaces.
3. Case design is simplified; an oiltight enclosure is not required.
4. The coupling may operate for extended periods of time without servicing except for normal leakage checks.
5. Seals are required in connected equipment.
6. Seals tend to wear and age, and replacement is not always easy.
7. High ambient temperatures may adversely affect the lubricant.

Continuous Lubrication

1. The coupling can be operated continuously, but sludge must be removed periodically.

2. Generated heat is removed effectively, and coupling life is significantly longer in applications subjected to high ambient temperatures.
3. Costly seals may not be needed in the connected equipment.
4. The coupling itself does not require a seal.
5. The choice of lubricating oil is limited.
6. The oil supply must be filtered to 5–10 μm or less absolute particle size.
7. An oiltight case is required.

D. Grease Lubrication

Because gear, chain, and grid couplings accommodate misalignment and transmit torque in similar ways, a grease that works satisfactorily in one should be good for the others. Various manufacturers have many recommendations for lubricants to use in their couplings, and it is very difficult to find a common denominator. Many tests and studies have been undertaken for gear couplings, and the conclusion should also apply to chain and grid couplings. The most important discovery of many of these studies is that the wear rate of a coupling is greatly influenced by the viscosity of the base oil of the grease—the higher the viscosity, the lower the wear rate.

One study showed that centrifugal force is not only important in helping to lubricate a coupling but can also cause grease to deteriorate and create serious problems. The sliding parts of a lubricated coupling would run dry if it were not for the centrifugal forces created by the rotation of the coupling. Centrifugal force is generated by the coupling's rotation and is a function of the coupling's diameter and the square of the rotational speed. Centrifugal force generates pressure in the lubricant. This pressure helps the coupling in two ways: It forces the lubricant to assume an annular form, flooding the coupling's teeth (or chain, or steel grid), and it forces the lubricant to fill all the voids rapidly.

Figure 4.36 can be used to calculate the magnitude of the centrifugal force g in a coupling. For example, for a 4-in. pitch diameter gear coupling at 3600 rpm, the centrifugal force can exceed $500g$. Because the centrifugal force is a function of the square of the rotational

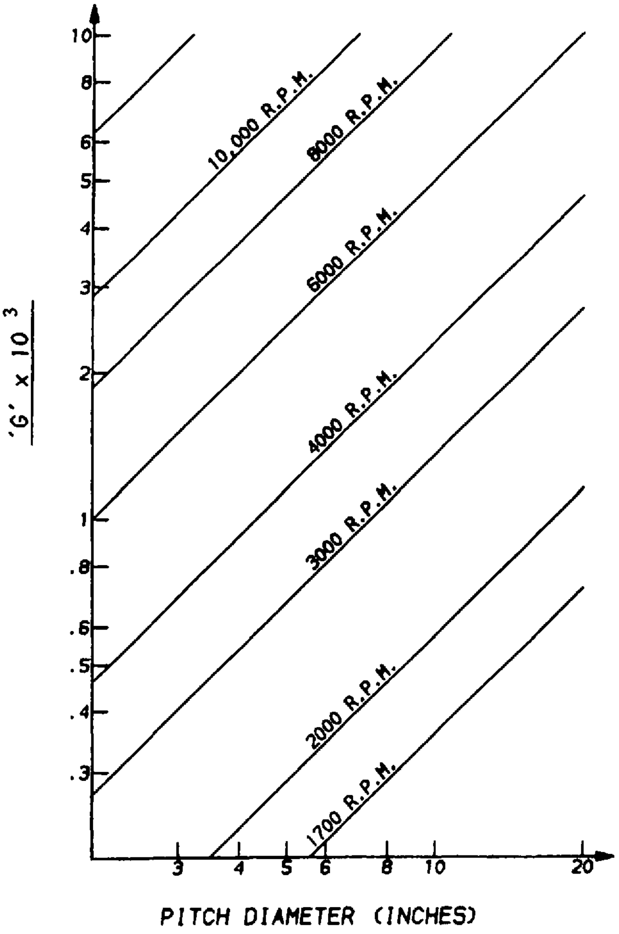


FIGURE 4.36 Centrifugal force developed in a coupling.

speed but the frequency of the hub tooth oscillatory motion increases in direct proportion to the speed, couplings get better lubrication when operating at high speeds. The reverse is true when a coupling operates at very low speeds The centrifugal force decreases rapidly and below a given level can no longer force a thick lubricant, such as grease, between the coupling teeth, so rapid wear is a result.

Lubricant selection depends on the type of coupling and particularly on the application. Even the best lubricant for one application can allow rapid wear in another application. This fact cannot be overemphasized. A coupling user who lubricates all the couplings in the plant with whatever grease is available cannot expect good performance from the couplings. In reality, a substantial reduction in maintenance costs can be realized through the use of proper lubricants.

The best approach is to read and follow the coupling manufacturer's recommendations. If lubricant recommendations are not available, the following can be used as a guide:

Steel grid coupling manufacturers generally recommend an NLGI No. 1 or 2 grease.

Gear and chain coupling manufacturers recommend an NLGI No. 0, 1, or 2 grease.

Normal applications can be defined as those where the centrifugal force does not exceed 200*g*. In general, these couplings can operate at motor speeds of 3600 rpm or less. Also, in normal applications the misalignment at each hub is less than 3/4° and the peak torque is less than 2 1/2 times the continuous torque. For these applications, an NLGI No. 2 grease with a high-viscosity base oil [preferably higher than 198 centistokes (cSt) at 40°C] should be used.

Low-speed applications can be defined as those where the centrifugal force is lower than 10*g*. If the pitch diameter (d) is not known, the following formula can be used:

$$\text{rpm} \leq \frac{200}{\sqrt{d}}$$

The same conditions for misalignment and shock torque as those above are valid. For these applications, an NLGI No. 0 or No. 1 grease with a high-viscosity base oil (preferably higher than 198 cSt at 40°C) should be used.

High-speed applications can be defined as those where the centrifugal force is higher than 200*g*, the misalignment at each hub is less than 1/2°, and the torque transmitted is fairly uniform. For these applications, the lubricant should have very good resistance to centrifugal separation. This information is seldom published but can be obtained from either the lubricant producer or the coupling manufacturer.

High-torque, high-misalignment applications can be defined as those where the centrifugal forces are lower than 200*g*, the misalignment is larger than 3/4° and the shock loads exceed 2.5 times the continuous torque. Many such applications also have high ambient temperatures (e.g., 100°C), at which only a few greases can perform satisfactorily. Besides having the characteristics of a grease for "normal applications," the grease should have antifriction and antiwear additives (such as molydisulfide), extreme pressure (EP) additives, a Timken load greater than 40 lb, and a minimum dropping point of 150°C.

E. Oil Lubrication

Oil is seldom used as a lubricant for chain or grid couplings but is very common for gear couplings. The couplings can be oil-filled or continuous, dipped or submerged. For oil-filled couplings the oil should be of a high-viscosity grade, not less than 150 SSU at 38°C. The higher the viscosity the better. For high-speed applications an oil viscosity of 2100–3600 SSU at 38°C can be used very successfully. Generally, an oil that conforms to MIL-L-2105 Grade 140 is acceptable. The only problems with an oil of such high viscosity is that it is very difficult to pour to fill a coupling, but the good operating characteristics obtained are usually well worth the time it takes.

Continuously lubricated couplings use the oil from the system, which is seldom a high-viscosity oil. Because of this, it is best to cool the lubricant before it enters the coupling. This will help supply the lubricant to the coupling at its highest lubricity (viscosity). The viscosity should be a minimum of 50 SUS at 100°C. An oil that conforms to MIL-L-17331 is usually satisfactory.

VI. COUPLING DISASSEMBLY

A. Opening the Coupling

To begin disassembly of a coupling it is advisable to proceed as follows:

1. Visually inspect the coupling before opening it. This can sometimes help in planning the required maintenance proce-

dure. For example, loose or missing bolts indicate poor installation practices; discolored (usually blue) coupling parts indicate that the coupling probably ran dry, and lubrication procedures should be reviewed. If sleeves, covers, or seal carriers have to be slid off the hubs, clean the exposed hub surface with emery paper to remove rust or corrosion. Examine elastomer parts for wear, which indicates that they came in contact with stationary components; they should also be examined for localized melting or permanent set and for signs of hardening or chemical attack. It is important to observe the state of a coupling even if it is damaged beyond repair. Understanding how or why a coupling failed can help to prevent similar occurrences in the future.

2. Observe the inside surfaces of coupling guards. This can provide clues to any coupling problems. The presence of grease streaks indicates not only that lubricant has leaked but also the area from which it was leaking. In the case of elastomer couplings, shredded elastomer material on the inside of a guard indicates possible contact between the element and the guard. Elastomer elements under centrifugal force grow bigger than when they are static.
3. For continuously lubricated gear couplings, verify whether the oil nozzles are properly positioned and whether the nozzles are clogged. If possible, start the oil pumps and measure the oil flow through each nozzle.
4. Check the ease of moving the coupling axially to verify whether it can accommodate shaft movement and whether it has become locked up.
5. Examine elastomer elements for chafing, which indicates that they came in contact with stationary components (such as the guards). Examine them for melted spots or signs of aging or chemical attack.
6. Balanced coupling components are match-marked. Verify that marks are still visible and if they are not, make new ones. Even if the coupling was component-balanced or not balanced at all, it is best to match-mark all major components so you can put the coupling back together the way it was. If the cou-

> pling was running well, it liked how it was assembled. The teeth of gear couplings tend to wear in after some time; therefore if the coupling is going to be reused it is best to matchmark it tooth to tooth and reassemble it the way it was.

Before opening a coupling it is advisable to have on hand a container in which to place the parts as you remove them. It is important to remember that most couplings use special bolts that should not be replaced with bolts of lesser quality, different grips, or different thread length. It is particularly important for some balanced couplings to check to see if there are match marks. If not and the coupling is to be reused, scribe the match marks *before* you open the coupling.

To shorten downtime, it is also best to have on hand all the parts that might need replacing, such as gaskets, O-rings, lubricant, and possibly a spare flexible element or an entire coupling.

After visual inspection, remove the bolts. Make sure that you use wrenches of the proper size; this is very important. Otherwise, the heads of the bolts or the nuts could be damaged. Do not use vise grips or adjustable wrenches. If lockwashers are used, it is better to loosen the bolt rather than the nut so as to keep the lockwasher from digging into the flange and nut surface.

Once the bolts have been removed, one would expect the coupling to open up easily, but this is seldom the case. Parts usually stick together, particularly when gaskets are used. To separate two flanges it is best to use the jacking holes (if there are any). Tapping the parts with a soft-face hammer can also help in separating parts. Prying the parts apart by hammering, chiseling, or prying with a screwdriver is not recommended, as it will damage the surfaces of the flanges, and the surfaces will be difficult to seal when reassembled.

Be prepared for a mess, as some of the lubricant (if it is a lubricated coupling) will leak out of the coupling when it is opened. All the lubricant should be removed from the coupling and discarded, as lubricant properties may deteriorate when a coupling is opened.

When the connected machines are removed from their location or when couplings are left disconnected for long periods of time, it is best to bolt a piece of cardboard or plywood to the half-coupling. There are two advantages to this procedure: The coupling is protected from dirt and contamination, and the coupling bolts that are used to hold the sleeves or cover will not get lost.

B. Removing the Hubs from the Shafts

Most installed hubs need some force to be removed. The hubs can be removed by applying a continuous force or through impact. Hammering will always damage the coupling, which is all right if the coupling is to be replaced. However, hammering is not recommended because it can also damage the bearing in the equipment and even bend shafts.

The first thing to do before removing the hub is to remove the locking means, such as tapered keys, setscrews, and intermediate bushings, on the shaft or nut. Make sure that all retaining rings, bolts, setscrews, and so on are loosened.

The most common way to remove the hubs is by using the puller holes (Figure 4.37). If the hubs have no puller holes when received, two holes can be drilled and tapped before the hub is installed on the shaft. It is best to buy hubs with puller holes equal to one-half the hub wall thickness to ensure proper sizing. To facilitate removal of the hub, use oil to lubricate the threads and use penetrating oil between the hub and the shaft. For straight shafts, use a spacer between the shaft end and the

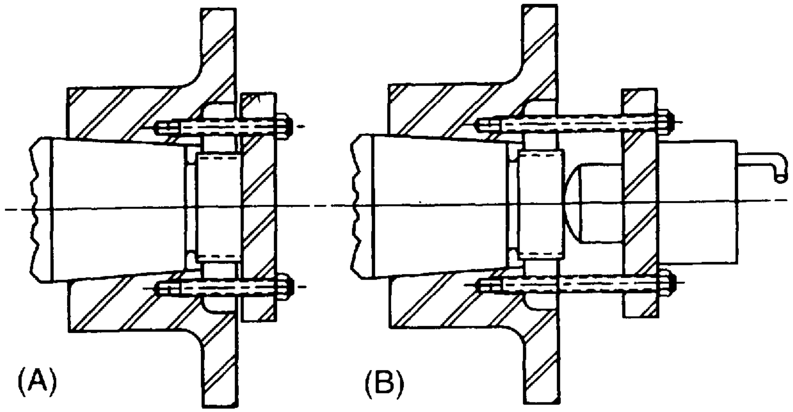


FIGURE 4.37 Methods of removing hubs. (A) Puller holes and pull-up plate. (B) Puller holes and puller press.

flat bar, as force has to be applied for as long as the hub slides on the shaft. If there are no puller holes, a "wheel puller" can be used; however, care should be taken not to damage the coupling. Figure 4.38 illustrates how not to pull on a gear coupling. Pulling on the sleeves of a coupling could damage it. Pulling from the back of the hub is sometimes a problem due to lack of room.

If the hub does not move even when maximum force is applied, apply a blow with a soft-face hammer. If nothing happens, apply heat to the hub. It is important to use a low-temperature flame (not a welding torch) and to apply the heat uniformly around the hub (localized temperature should not exceed 600°F for most steel). Heat does not have the same effect in this case as it has at installation, when only the hub is heated. At removal, the hub is on the shaft, and the heat will also expand the shaft. The secret is to heat only the outside diameter of the hub, shielding the shaft from the heat. If possible, wrap the shaft with wet rags. Apply the heat *while the pulling force is being applied.*

Caution: If heat has been applied to remove the hub, hardness

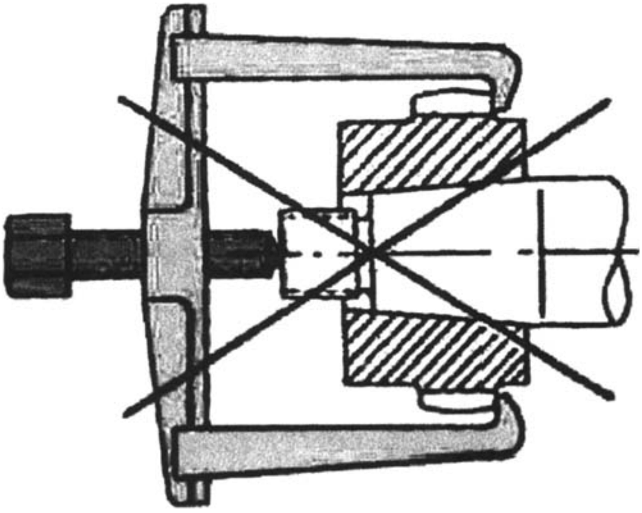


FIGURE 4.38 Improper way to remove hubs with a wheel puller.

checks should be made at various sections before using the hub again, to ensure that softening of the material has not occurred. If there is any doubt, the coupling should be replaced. Rubber components should be removed during heating. If they are not removed, they will have to be replaced if they come into contact with excessive heat.

The force required to remove a hub is a function of its size (bore and length) and of the interference used at installation. For very large couplings, pressure can be applied in two ways for hub removal. The first is to use a portable hydraulic ram such as the puller press shown in Figure 4.37B; the second is to use a pull-up plate, as in Figure 4.37A. The force that can be applied is limited only by the strength of the threads and rods.

There are many cases where nothing helps to remove the hub; then the hub has to be cut. This is usually cheaper than replacing the *shaft*. The place to cut is above the key, as shown in Figure 4.39B. The cut should be made with a saw, but a skilled welder could also flame-cut the hub, taking precautions that the flame does not touch the shaft. After the cut is made, a chisel is hammered into the cut (Figure 4.39B), spreading the hub and relieving its grip on the shaft. If the hub is too thick, it can be machined down to a thin ring before it is split (Figure 4.39A).

If keyed hubs are to be removed often, they can be designed so they can be removed hydraulically. A groove is machined into the middle of the hub bore in such a way that the ends of the groove do not enter the keyway. A hole is drilled to connect the groove to an outside oil fitting. Oil is pumped into the hub, and the hub can be removed with the aid of a hydraulic puller.

C. Loosening Intermediate Bushings

To loosen intermediate bushings, remove the bolts (usually more than two) and insert them in the alternate holes provided, making sure to lubricate the threads and the bolt (or setscrew) points. When tightened, the bolts will push the bushing out of the hub and relieve the grip on the shaft. The bushing does not have to be moved more than 1/4 in. If the bushing is still tight on the shaft, insert a wedge in the bushing's slot and spread it open. Squirting penetrating oil into the slot will also help in sliding the hub–bushing assembly off the shaft. Do not spread the bush-

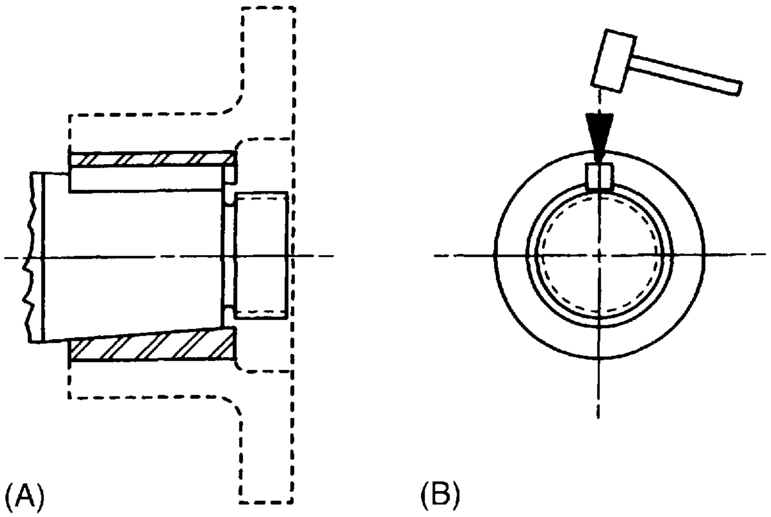


Figure 4.39 Splitting a hub to remove it from a shaft. (A) Turn a thick hub down when possible. (B) Split over the keyway.

ing open without the hub on it, as this will result in the bushing breaking or yielding at or near the keyway.

D. Returning Parts to Service

Any part that is to be reused should first be thoroughly cleaned and examined. The part should be dimensionally inspected to verify that it is within the coupling manufacturer's limits. The part should be examined for dents, burrs, and corrosion. It should be nondestructively tested (magnaflux or dye penetration tested) for surface cracks or other signs of distress. If there is any question about the coupling's integrity, *throw it* away and use a new one or return it to the coupling manufacturer to determine its suitability for further service. Using damaged or deteriorated parts could result in serious failure of the coupling or the equipment.

5

Rigid Couplings

I. INTRODUCTION

Whenever two pieces of rotating equipment need to be connected, the question arises as to whether they should be directly coupled or indirectly coupled. Equipment can be indirectly coupled by belts or chains (see Figure 5.1A), but indirectly coupled equipment is usually less efficient than directly coupled equipment. This is normally due to the frictional losses from the belts and chains slipping and or sliding when they transmit power. Therefore, it is usually preferable to directly couple two pieces of rotating equipment (Figure 5.1B).

The main problem associated with directly coupled equipment is how to accommodate for the usual misalignment that can occur between two pieces of rotating equipment. Rigid couplings are used when misalignment does not exist or when it is very small or when the equipment is robust enough to handle the reactionary load produced from trying to flex rigid couplings. In those cases, rigid couplings are a very effective

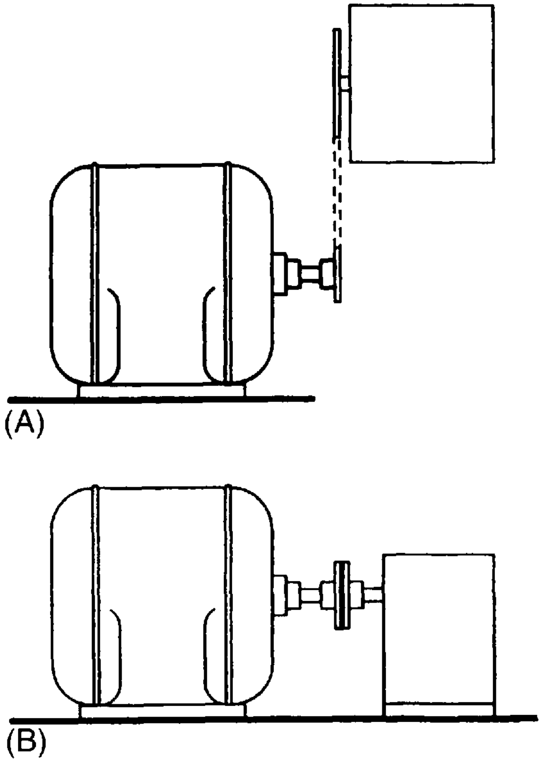


FIGURE 5.1 Equipment connection methods. (A) Indirectly coupled equipment. (B) Directly coupled equipment.

means of connecting equipment. The four basic types of rigid couplings have been illustrated in Chapter 2:

1. The flanged rigid coupling (Figure 2.7)
2. The ribbed rigid coupling (Figure 2.8)
3. The sleeve rigid coupling (Figure 2.9)
4. The quill shaft rigid coupling (Figure 2.10)

In general, rigid couplings connect two pieces of rotating equipment. They allow for the transfer of power from one piece of equipment to the other. They also allow equipment that has shafts of different sizes to be connected. It is important to stress the fact that rigid couplings should be used only when the equipment has virtually no misalignment and/or when the shafts of the equipment or the rigid coupling (quill shaft rigid coupling) are long and slender enough that they can flex and accept the forces and moments produced by the mechanical deflection on these parts due to the misalignment imposed by the connected equipment.

II. FLANGED RIGID COUPLINGS

The flanged rigid coupling is probably the most common type of rigid connection. It is available from almost every flexible coupling manufacturer in several standard series. One of the most common is interchangeable with standard flanged gear couplings. Figure 5.2 illustrates and lists some of their dimensions.

Rigid couplings are usually made from carbon steel 1035 to 1050 either from bar stock or from forgings. They can handle large amounts of torque for their sizes; Figure 5.2C also shows the typical torques that can be handled by the various sizes.

Many other configurations and sizes are available (see Figure 5.3). They can be supplied with replaceable bushings (see Figure 5.4); this allows the rigid coupling to be easily removed so that equipment bearings and seals can be replaced or serviced. The flanged rigid coupling is also available in various materials: gray iron, malleable iron, various carbon steels, and alloy steels. The design is usually limited by the number, size, and type of bolts used (see Section II of Chapter 3) for the various stress calculations that are usually considered. The ratings should consider the stresses in bolts, hubs, and the flange itself. It is suggested that you check with specific coupling manufacturers for information on material used, bore sizes allowed, number of bolts used, and torque and speed ratings.

The flanged rigid coupling can be used wherever no misalignment is present, or virtually none. Some applications include pumps (vertical and horizontal) and crane drives (see Figure 5.5).

III. RIBBED RIGID COUPLINGS

The ribbed rigid coupling is used where ease of assembly and disassembly is required. The coupling clamps onto the shaft. The shaft and

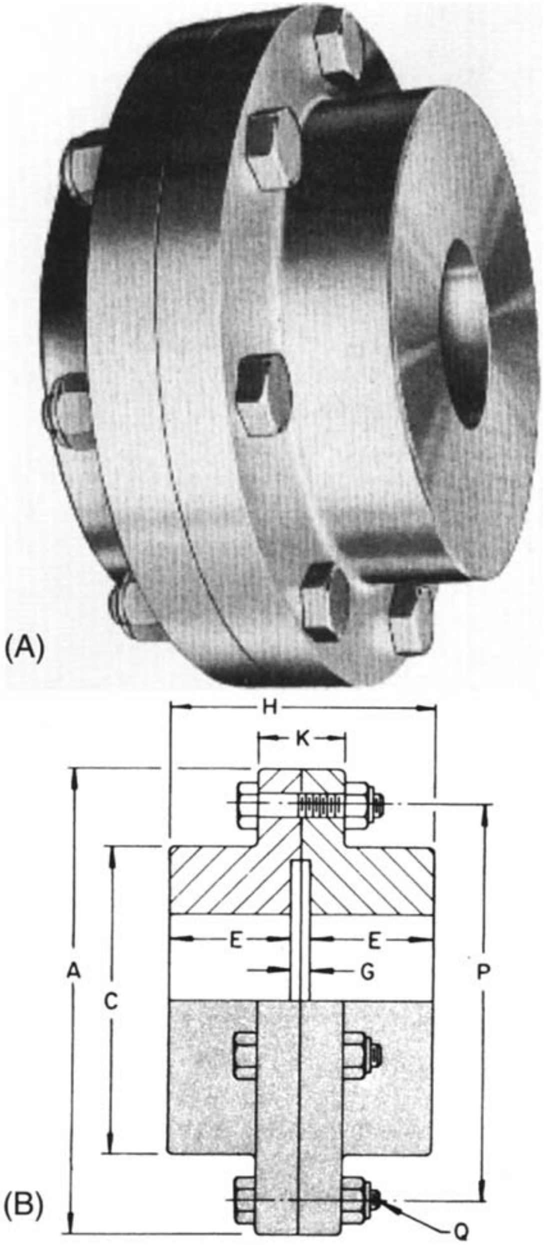
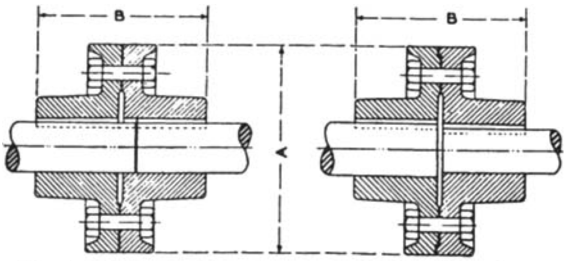


Figure 5.2 (A) Typical rigid coupling and (B) its dimensions. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

Size	Max. rpm	Torque (lb.-in.)	*A* diameter	*C* diameter	*E*	*G*	*H*	*K*	*P* bolt circle	*Q* bolts	
										Number	Size
1W	10,000	8820	$4\frac{9}{16}$	3	$1\frac{9}{16}$	$\frac{3}{16}$	$3\frac{9}{32}$	$1\frac{1}{8}$	$3\frac{3}{4}$	6	$\frac{1}{4}$–28 × $1\frac{1}{2}$
$1\frac{1}{2}$W	7,400	22,680	6	$3\frac{13}{16}$	$1\frac{27}{32}$	$\frac{3}{16}$	$3\frac{27}{32}$	$1\frac{1}{2}$	$4\frac{13}{16}$	8	$\frac{3}{8}$–24 × 2
2W	5,900	44,100	7	$4\frac{13}{16}$	$2\frac{9}{32}$	$\frac{3}{16}$	$4\frac{23}{32}$	$1\frac{1}{2}$	$5\frac{7}{8}$	6	$\frac{1}{2}$–20 × $2\frac{1}{4}$
$2\frac{1}{2}$W	5,000	75,600	$8\frac{3}{8}$	$5\frac{23}{32}$	$2\frac{29}{32}$	$\frac{3}{16}$	6	$1\frac{7}{8}$	$7\frac{1}{8}$	6	$\frac{5}{8}$–18 × $2\frac{3}{4}$
3W	4,300	132,300	$9\frac{7}{16}$	$6\frac{23}{32}$	$3\frac{13}{32}$	$\frac{3}{16}$	7	$1\frac{7}{8}$	$8\frac{1}{8}$	8	$\frac{5}{8}$–18 × $2\frac{3}{4}$
$3\frac{1}{2}$W	3,900	189,000	11	$7\frac{3}{4}$	$3\frac{31}{32}$	$\frac{3}{16}$	$8\frac{5}{32}$	$2\frac{1}{4}$	$9\frac{1}{2}$	8	$\frac{3}{4}$–16 × $3\frac{1}{4}$
4W	3,500	299,250	$12\frac{1}{2}$	$8\frac{31}{32}$	$4\frac{7}{16}$	$\frac{3}{8}$	$9\frac{3}{16}$	$2\frac{1}{4}$	11	8	$\frac{3}{4}$–16 × $3\frac{1}{4}$
$4\frac{1}{2}$W	3,200	409,500	$13\frac{5}{8}$	$10\frac{1}{8}$	5	$\frac{3}{8}$	$10\frac{11}{32}$	$2\frac{3}{14}$	12	10	$\frac{3}{4}$–16 × $3\frac{1}{4}$
5W	2,900	576,450	$15\frac{5}{16}$	$11\frac{3}{8}$	$5\frac{3}{4}$	$\frac{3}{8}$	$11\frac{27}{32}$	3	$13\frac{1}{2}$	8	$\frac{7}{8}$–14 × $4\frac{1}{4}$
$5\frac{1}{2}$W	2,700	756,000	$16\frac{9}{16}$	$12\frac{9}{16}$	$6\frac{1}{8}$	$\frac{3}{8}$	$12\frac{19}{32}$	3	$14\frac{1}{2}$	14	$\frac{7}{8}$–14 × $4\frac{1}{4}$
6W	2,500	1,008,000	18	$13\frac{7}{8}$	$7\frac{1}{4}$	$\frac{3}{8}$	$14\frac{27}{32}$	2	$15\frac{3}{4}$	14	$\frac{7}{8}$–14 × $3\frac{1}{4}$
7W	2,200	1,575,000	$20\frac{3}{4}$	$15\frac{3}{4}$	$8\frac{11}{16}$	$\frac{1}{2}$	$17\frac{13}{16}$	$2\frac{1}{4}$	$18\frac{1}{4}$	16	1–14 × $3\frac{1}{2}$

FIGURE 5.2 (C) List of dimensions and torques.



Standard Plain Face Type

Male and Female Reducing Coupling

Shaft Size	Included Sizes	Weight lbs.	A	B	Shaft Size	Included Sizes	Weight lbs.	A	B
1 3/16	1 1/4	15	6	4 5/8	3 15/16	4	130	12 1/2	10 5/8
1 7/16	1 1/2	19	6 3/4	5 1/8	4 7/16	4 1/2	181	13 1/2	13 5/8
1 11/16	1 3/4	26	7 1/4	5 3/4	4 15/16	5	251	16 1/4	14 3/8
1 15/16	2	34	8	6 3/8	5 7/16	5 1/2	310	17 1/2	15 1/8
2 3/16	2 1/4	40	8 1/2	6 7/8	5 15/16	6	400	19	16 1/8
2 7/16	2 1/2	48	9	7 3/8	6 7/16	6 1/2	525	20.	16 3/4
2 11/16	2 3/4	57	9 3/4	7 7/8	6 15/16	7	575	21	17 1/2
2 15/16	3	74	10 1/2	8 3/8	7 7/16	7 1/2	800	22 1/2	18 1/2
3 7/16	3 1/2	93	11 1/4	9 3/8	7 15/16	8	875	24	19 1/2

FIGURE 5.3 Other rigid coupling configurations. (Courtesy of Dodge Division of Reliance Electric.)

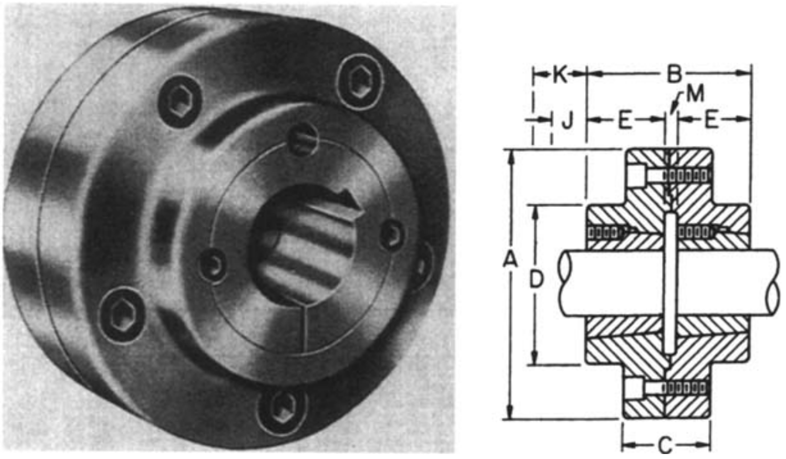


FIGURE 5.4 Rigid couplings with replaceable bushings. (Courtesy of Dodge Division of Reliance Electric.)

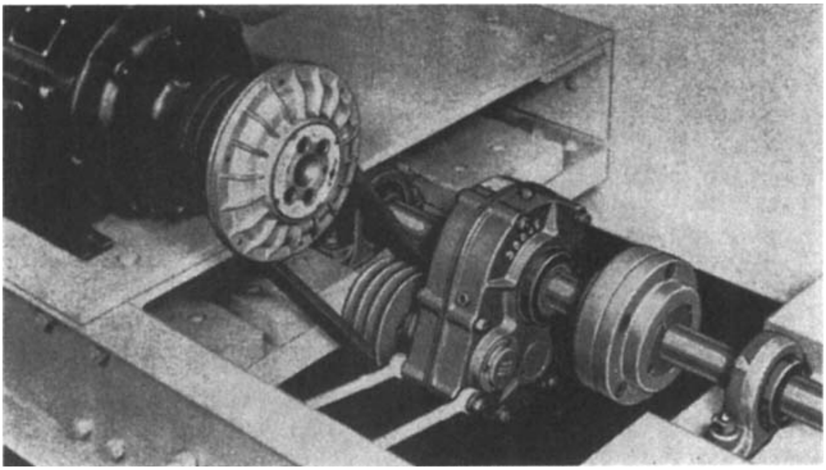


Figure 5.5 Shop crane application of a flanged rigid coupling. (Courtesy of Dodge Division of Reliance Electric.)

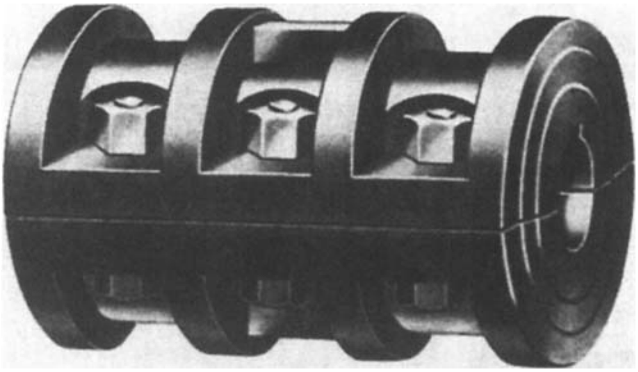
the coupling hubs are usually keyed. The two halves are held together by radial bolts at the split. The quantity of bolts can vary depending on the size of the coupling; typically, four to eight are used. Torque is transferred from one half to the other by the frictional force produced by the bolts rather than by direct loading of the bolts.

These couplings are usually made of material having a strength compatible with AISI 1018 steel shafts (relatively low torque capacities). Many coupling manufacturers make a standard line of ribbed couplings. There is no industrial standard. Typically, these couplings are available in bore sizes from 1 to 7 in. Figure 5.6 provides some typical information and dimensions.

Ribbed rigid couplings are usually used for low-speed, low-torque applications. They are typically used on vertical pumps, agitators, and winch drives (see Figure 5.7) and in many other types of applications.

IV. SLEEVE RIGID COUPLINGS

The sleeve rigid coupling is one of the simplest forms of coupling available. On small (usually fractional horsepower) drives where the equip-



Shaft size	Maximum bore	Maximum rpm	Outside diameter	Length	Number and size of bolts
1, 1 3/16, 1 1/4	1 1/4	5360	3 5/8	5 1/4	4–3/8
1 3/8, 1 7/16, 1 1/2	1 1/2	4130	4 5/8	6 3/16	4–1/2
1 11/16, 1 3/4	1 3/4	3965	4 13/16	7 1/16	4–1/2
1 7/8, 1 15/16, 2	2	3635	5 1/4	7 15/16	4–1/2
2 3/16, 2 1/4	2 1/4	3180	6	8 5/8	4–5/8
2 7/16, 2 1/2	2 1/2	2965	6 7/16	9 11/16	6–5/8
2 11/16, 2 3/4	2 3/4	2830	6 3/4	10 9/16	6–5/8
2 15/16, 3	3	2545	7 1/2	11 3/8	6–3/4
3 3/16	3 1/4	2315	8 1/4	12 1/4	6–7/8
3 7/16	3 1/2	2165	8 13/16	13 3/16	6–7/8
3 15/16	4	1900	10 1/16	15 1/4	6–1
4 7/16	4 1/2	1775	10 3/4	18 3/16	6–1 1/8
4 15/16	5	1625	11 3/4	19 5/8	6–1 1/8
5 7/16	5 1/2	1390	13 3/4	20 3/8	8–1 1/8
5 15/16	6	1365	14	20 3/4	6–1 1/4
7	7 1/16	1230	15 1/2	21 15/16	8–1 1/4

Figure 5.6 Ribbed rigid coupling: typical dimensions. (Courtesy of Dodge Division of Reliance Electric.)

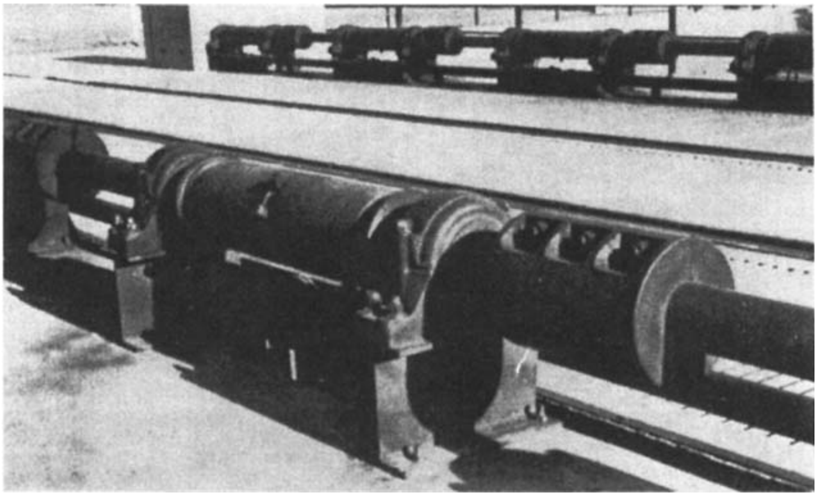
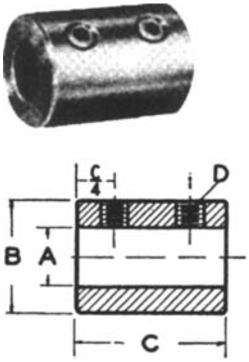


FIGURE 5.7 Ribbed rigid coupling application. (Courtesy of Dodge Division of Reliance Electric.)

ment shafts are of the same diameter, a sleeve rigid coupling can be slid onto the shaft of one piece of equipment, the equipment put in place, and the sleeve rigid coupling slid onto the shaft of the other piece of equipment. The sleeve is usually locked to the shafts with two setscrews, one for each piece of equipment. What could be simpler?

There is no industrial standard for this type of coupling. Figure 5.8 provides some information for a standard line offered by one coupling manufacturer. For larger sizes the sleeve rigid coupling can be supplied with a replaceable bushing for ease of assembly and disassembly. Figure 5.9 lists some typical dimensions.

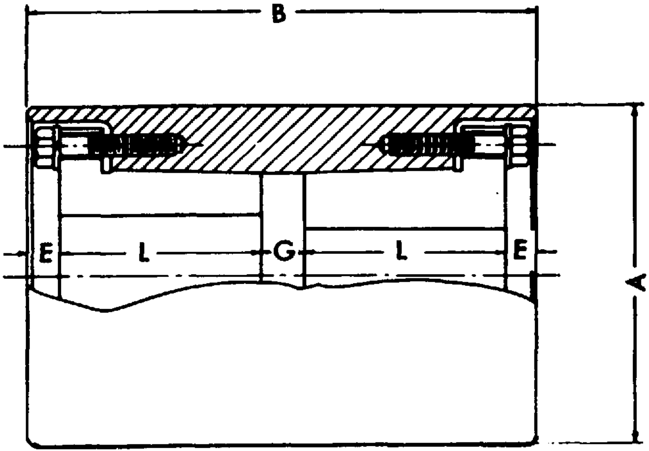
For high-torque applications a more sophisticated sleeve rigid coupling has been used. The coupling consists of two sleeves of high-quality steel (see Figure 5.10), a thin inner sleeve and a thick outer sleeve. The outer surface of the inner sleeve is tapered, and the bore of the outer sleeve has a corresponding taper. The inner sleeve bore is somewhat larger than the diameter of the shafts so that the sleeve can be passed over them with ease. The outer sleeve is then driven up the



Part number	B (in.)	C (in.)	A (in.)	C/4 (in.)	Weight
CS—04	1/2	3/4	1/4	3/16	0.06
CS—05	5/8	1	5/16	1/4	0.06
CS—06	3/4	1	3/8	1/4	0.1
CS—08	1	1 1/2	1/2	3/8	0.2
CS—10	1 1/4	2	5/8	1/2	0.5
CS—12	1 1/2	2	3/4	1/2	0.8
CS—14	1 3/4	2	7/8	1/2	1.0
CS—16	2	3	1	3/4	1.9
CS—18	2 1/8	3	1 1/8	3/4	2.1
CS—20	2 1/4	4	1 1/4	1	3.1
CS—22	2 1/2	4 1/2	1 3/8	1	4.3

FIGURE 5.8 Sleeve rigid coupling: typical dimensions.

tapered inner sleeve using the hydraulic unit incorporated in the coupling. This action compresses the inner sleeve onto both shafts. To allow this drive-up, the friction of the matching tapered surfaces is first overcome by injecting oil at high pressure between them, where it forms a load-carrying film separating the two components. When the outer sleeve has reached its correct position, the injection pressure is released and the oil is drained off, restoring normal friction between the sleeves.



(A)

Coupling number	Maximum bore		Bushing	Dimensions					Weight (including bushings)
	Light loads	Heavy loads		A	B	E	G	L	
44-SD	1 3/4	1 1/2	SD	4	4 5/8	3/8	1/4	1 13/16	11
44-SF	2 1/2	2 1/4	SF	5 1/2	5 1/4	1/2	1/4	2	22
44-E	3 3/16	2 3/4	E	6 7/8	6 3/4	5/8	1/4	2 5/8	54
44-J	4	3 5/8	J	8 1/4	11	3/4	1/2	4 1/2	122
44-M	5 1/4	4 11/16	M	10	16	1	1/2	6 3/4	270

(B)

Figure 5.9 (A) Sleeve rigid coupling with replaceable bushings. (B) Data on the coupling. (Courtesy of TB Wood's Inc.)

Dismounting the coupling is equally simple. Oil is injected between the coupling sleeves to overcome the friction. As a result of the taper, the compressive force has an axial component that causes the outer sleeve to slide down the taper, forcing the oil out of the hydraulic unit. By controlling the flow of this oil, the sleeve can be prevented from sliding too quickly.

Sleeve rigid couplings with setscrews are used on motor-driven pumps. The hydraulic sleeve rigid coupling is used on high-torque applications such as marine propulsion shafting (see Figure 5.11).

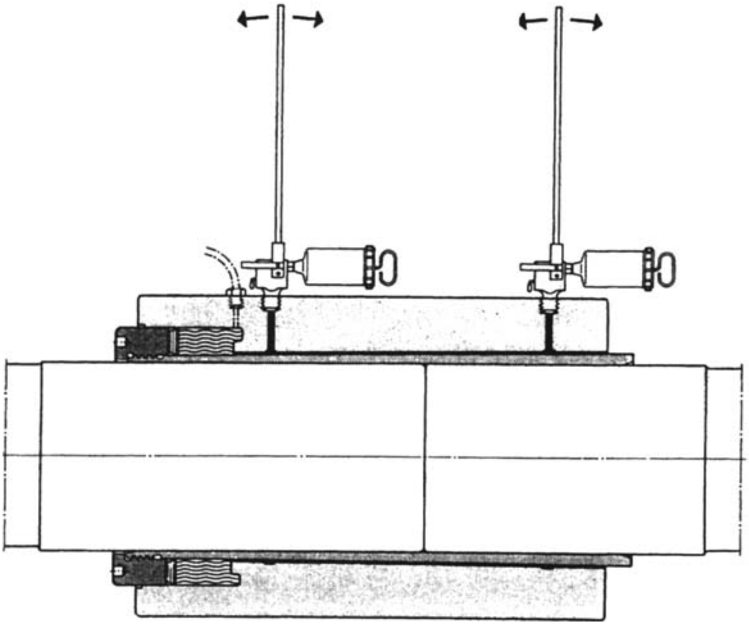


FIGURE 5.10 High-torque sleeve rigid coupling. (Courtesy of SKF Steel, Coupling Division.)

V. QUILL SHAFT RIGID COUPLINGS

The quill shaft rigid coupling gets its name from the fact that it looks like the quill shafts that go through some large gears in gearboxes; they are usually long and slender. These rigid couplings do accommodate for some misalignment; they do this through the flexing of their long slender shafts. Quill shaft rigid couplings are usually much smaller than flexible couplings, generally 25–50% smaller. For example, a 40 in. quill shaft rigid coupling is approximately equivalent to a 60 in. flexible coupling.

Quill shaft rigid couplings are usually made of high-grade alloy steels. This is because the stresses imposed on them when they are misaligned (flexed) are usually quite high and they must be designed for the cyclic loads imposed on them when they are flexed due to misalignment. These couplings are usually used on large, high-horsepower steam and gas turbines (see Figure 5.12).

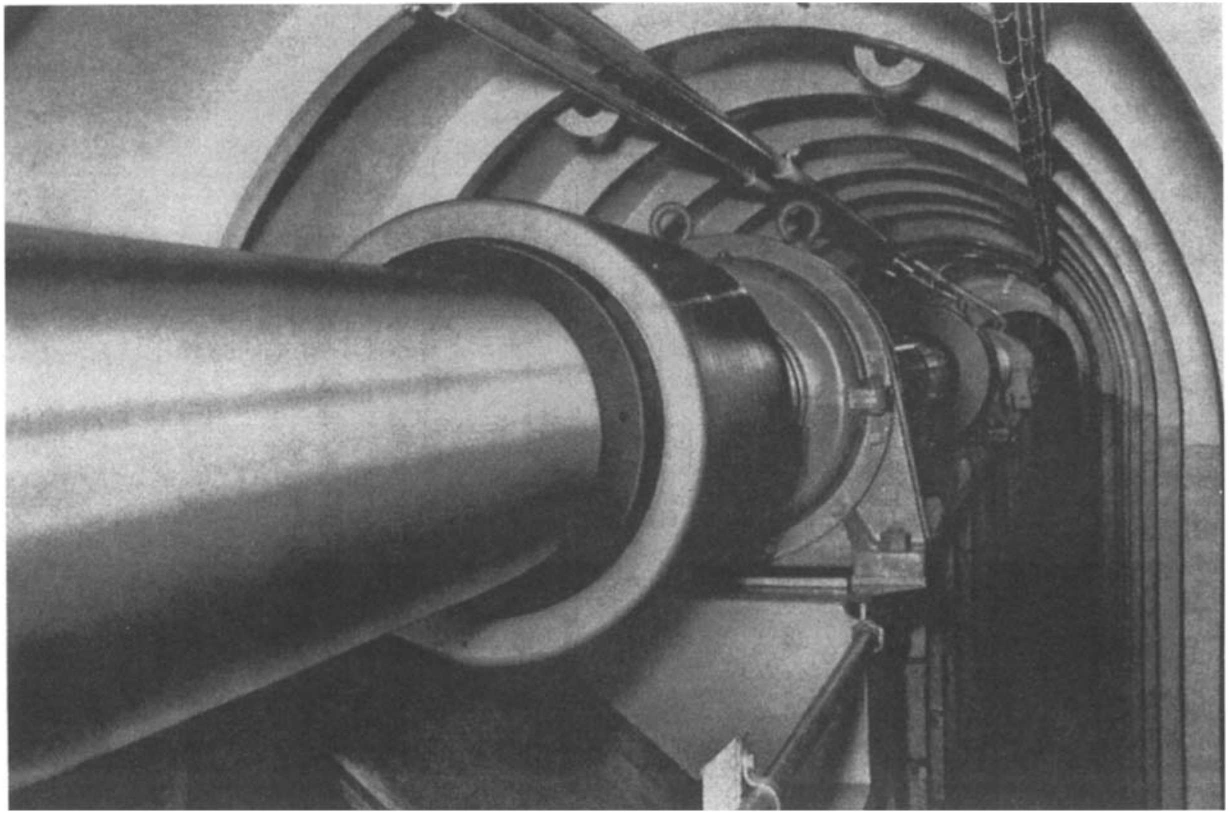


Figure 5.11 Application of a sleeve rigid coupling for a propeller shaft. (Courtesy of SKF Steel, Coupling Division.)

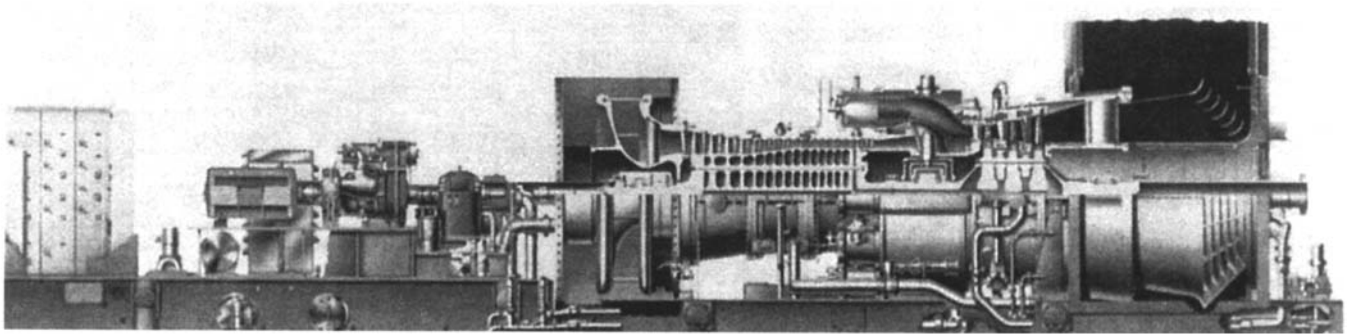


FIGURE 5.12 Quill shaft rigid coupling. (Courtesy of General Electric Schenectady, Gas Turbine Division.)

6

Mechanically Flexible Couplings

I. INTRODUCTION

The mechanically flexible coupling meets the three basic requirements of a flexible coupling: it transmits power, accommodates for misalignment, and compensates for shaft end float. The couplings in this group obtain their flexibility from loose-fitting parts and/or the rolling or sliding of mating parts. They usually require lubrication unless one of the moving parts is made of a material that supplies its own lubrication, such as nylon (chain-on-chain couplings and sleeves-on-gear couplings).

There are four basic types of couplings in this group:

1. The gear coupling
2. The chain coupling
3. The grid coupling
4. The universal joint

Gear couplings have been applied to bores to 36 in., misalignment to $7\frac{1}{2}°$, speeds to 40,000 rpm, and torques to 40,000,000 in.-lb. *Chain couplings* have been applied to bores to 10.5 in., misalignment to 2°, speeds to 6500 rpm, and torques to 1,350,000 in.-lb. *Grid couplings* have been applied to bores to 20 in., misalignment to $\frac{1}{3}°$, speeds to 4000 rpm, and torques to 4,000,000 in.-lb. *Universal joints* have been applied to bores to 20 in., misalignments to 15°, speeds to 8000 rpm, and torques to 25,000,000 in.-lb.

Note that the foregoing information is based on presently published information. It is unlikely that all the maximum conditions could be handled in any or all of the combinations listed. The couplings in this group are some of the most commonly used in the industry. Although these couplings must be lubricated, when they have been properly installed, lubricated, and maintained, they have been known to last 25 years or more. Normally, the life span is 3 to 5 years. The most common mode of failure for this group of couplings is that they wear out. This type of failure mode is usually predictable and detectable at maintenance checks and/or from the symptoms associated with a roughly running piece of equipment. This type of failure mode usually allows for replacement of the coupling before a disastrous failure occurs. This does not preclude other failure modes, such as bolt breakage, hub splitting, spacer breakage, or other broken parts, but wear-out of flexing elements is the most common. Mechanically flexible coupling failures can usually be attributed to sloppy installation, poor lubrication, and inadequate maintenance practices. In the following section we discuss in detail how some of the mechanical flexible couplings work, where they are used, various types, what is considered in their design, how they affect a system, and failure modes.

II. GEAR COUPLINGS*

A. Variables

A gear coupling is one of the simplest and most common types of couplings in use today. It is also one of the most difficult to design and evaluate, because of the number of variables that can affect its successful operation. Some of these variables are:

---

*Figures for this section are grouped together starting on p. 289.

1. Tooth design
    a. Straight teeth
    b. Type of crown and amount of crown
    c. Pressure angle of tooth
    d. Amount of backlash
    e. Accuracy of tooth spacing
2. Material
    a. Medium-carbon steel, alloy steels
    b. Type of core heat treatment: as-rolled, normalized, heat-treated
    c. Type of surface hardening: none, induction, nitrided, carburized
3. Lubrication
    a. Oil
    b. Grease
    c. Sealed lubrication
    d. Continuous lubrication

These are just some of the important variables and indicate why no general criteria have been established. Experience and success are the real criteria and the proof of a design. Section II of Chapter 3 gives a method of comparing various stresses in coupling components. Section F of this chapter gives a method of comparing the relative life, of gear couplings.

B. How Gear Couplings Work

A gear coupling usually consists of four major components, as illustrated in Figure 6.1. When two shafts are to be connected by this type of coupling, a coupling half (consisting of one hub and its mating sleeve) is keyed to each shaft, the shaft ends are brought together, and the sleeves are bolted together securely. Since the hubs have external teeth that mesh with the internal teeth on the sleeves, the connection is in effect an internal spur gear drive where the gear ratio is 1:1. When a coupling is misaligned (pivot position) with no load, only two teeth 90° from the plane of misalignment (tilted position) come into contact (see Figure 6.2).

Figure 6.1B indicates that there is a gap between the two hubs and that the sleeve teeth are somewhat longer than the hub teeth. It is therefore possible for axial movement of the hubs to occur without interfering with the operation of the coupling. If axial movement were the only type of misalignment encountered, the design of flexible couplings would be a relatively simple task. However, because perfect initial alignment of machinery is difficult to achieve and operating conditions impose forces on the shafts that tend to throw them out of line radially and angularly, it is necessary for the coupling to compensate for misalignments other than just those produced by axial movement.

Pure angular misalignment is illustrated in Figure 6.3A. The misalignment between shafts A and B is designated by the angle α. Figure 6.3B shows the case where the shafts are displaced radially. This type of misalignment may be resolved into pure angular misalignment, as in Figure 6.3A, by use of a floating shaft E between shafts C and D. Thus it can be seen that the two types of misalignments shown in Figure 6.3 are basically the same. Henceforth, the term *misalignment* will be applied to angular displacement and will be designated by the angle α.

To accommodate for misalignment, it is necessary to build considerable backlash into the teeth of the coupling. This is usually done by making the sleeve tooth thinner than its standard thickness. However, if the teeth on both the hub and sleeve are of conventional form, the teeth will be loaded at their end during misalignment. This is illustrated in Figure 6.4, which shows the entire load being applied at points P and Q. The tooth is loaded at these points during operation. At high loads and misalignments this may be an undesirable condition, as the ends of the hub teeth could dig into the faces of the sleeve teeth, and high bending moments would be developed at the point of contact because of the high coefficient of friction. One method of relieving end loading is to crown the teeth on the hub. The resultant form, shown in Figure 6.5A, may be compared with the standard tooth form shown in Figure 6.5B. Cutting a crowned tooth is done readily on a hobbing and shaping machine. It is a relatively simple matter to put a cam in the machine that will make the hob or shaper travel through a curved path rather than a straight line as it advances across the gear. The path of the hob and shaper is represented by the arrows in Figure 6.5.

The effect of crowning is illustrated in Figure 6.6. Figure 6.6A shows a coupling connecting two shafts whose angular misalignment is represented by the angle α. The relative position of the hub teeth and sleeve teeth at point *A* is shown in Figure 6.6B. Figure 6.6C shows the same for the teeth at point *B*. By comparing Figure 6.6C with Figure 6.4, it may be seen that the end-loading condition has been minimized. At maximum designed angle the loading would be at the ends. At less than maximum designed angle, the point of application of load is closer to the center of the tooth, and therefore rotationary moments are less. It may also be seen by inspection of Figure 6.6C that a crowned hub tooth allows more misalignment than a straight tooth because the ends (where contact would first occur) are rounded off so that the sleeve tooth in effect pivots about point *C*.

C. Where Gear Couplings Are Used

Gear couplings are used on the following types of equipment: centrifugal pumps, conveyors, exciters, fans, generators, blowers, mixers, hydraulic pumps, compressors, steel mills and auxiliary equipment, cranes, hoists, mining machinery, and so on. Figure 6.7 shows a compact hot rolling mill. This consist of a continuous caster, one rough standing, and six finishing stands. Figure 6.8 shows a typical configuration of one the finishing stands. There are three types of gear couplings used in primary rolling stands:

1. Figure 6.9 shows a size 15 alloy shear spacer coupling used between the motor and gear box—5600 kW @ 104/304 rpm, torque 4,550,000 lb-in.
2. Figure 6.10 shows a size 22 alloy spacer coupling used between the gear and pinion—5600 kW @ 23.4/68.5 rpm, torque 20,210,000 lb-in.
3. Figure 6.11 shows a size 11.2 carburized (with ground teeth) spindle that telescopes to accommodate approximately ± 5 in. for roll shifting. Operating angles are approximately 2°. 5600 kW @ 23.4 rpm (split 60%), torque 12,136,000 lb-in.

Figure 6.12 shows a Poole coupling of approximately 80 in. OD, weight 60,000 lb, with a 28-in.-diameter shaft, in a steel mill drive rated at 11,425,000 in.-lb at 19.3 rpm. Figure 6.13 shows a 25-ton 100-ft-tall yard crane. Figure 6.14 shows a typical gear coupling/pump application. Figure 6.15 shows a typical gear coupling/fan application.

D. Types of Gear Couplings

Gear couplings can be classified in many ways. Two of the most common are by type of tooth form and by functionality.

1. Type by Tooth Form

In all gear couplings the internal teeth are straight-sided teeth. They are involute formed teeth, usually at a 20° or 25° pressure angle (see Figure 6.16). Generally, most of backlash required to misalign a gear coupling is cut into the internal tooth.

a. Hub Tooth Form

1. *Straight hub tooth form:* This is the same form as that of the internal teeth described above (see Figure 6.17A).
2. *Crowned tooth with pitch diameter on a curve:* These teeth are crowned hobbed or shaped with a cutter on a cam. This produces a pitch diameter that is on a curve (see Figure 6.17B).
3. *Crowned tooth with pitch diameter on a straight line:* The external teeth are straight hobbed or shaped; then a crown is shaved on the tooth (see Figure 6.17C).
4. Variable crowned teeth: The external teeth are generated with a cam that has multiple curvatures (see Figure 6. 17D).

b. Which Tooth Form Is Better? It is difficult to say whether a crown is better than a straight tooth, and if it is, which crown, given equal pitch diameter, diametral pitch, and tooth spacing, since gear couplings rely mostly on tooth deflection to carry load. At small angles, of 1/4° or less, the straight tooth coupling works as well as the crowned tooth coupling. For small angles a tooth shaved or relieved by 0.0015 to 0.003 in. works like a hobbed or shaped crowned tooth. For small angles it is also probable that a straight tooth will wear itself into its required crown (for teeth not surface-hardened, nitrided, carburized, induction-hardened, etc.). Over 1/4°, all crowned teeth (curvatures of over 0.003 in.) work about the same, assuming that we are comparing the same curvatures produced by different methods.

2. Type by Functionality

Typing by function probably the best way to classify gear couplings. There are five functional classes for gear couplings:

1. The general-purpose gear coupling
2. The special-purpose high-speed gear coupling
3. The high-torque/high angle/low-speed gear coupling
4. The high-angle/high-speed gear coupling
5. Gear couplings for special applications

a. General-Purpose Gear Couplings There are two basic designs in the general-purpose group: straight tooth couplings (Figure 6.18) and crowned tooth couplings (Figure 6.19). Guston Fast was the inventor of this coupling and his basic design is still available from Kop-Flex (Fast's) and from TB Wood's (Poole). The general-purpose gear coupling is one of the few couplings that has some degree of standardization. Basically, flange, OD, and bolt circles are standard and most half couplings are interchangeable. Torques and bore capacities are not standard. Figure 6.20 provides a comparison of most commercial gear couplings. General-purpose gear couplings are also available in larger sizes, but standards and Interchangeability data do not exist for these sizes. Figure 6.21 shows one manufacturer's typical dimensions for these larger sizes. There are many variations of the general-purpose gear coupling, including the following:

Gear coupling with spacer (Figure 6.22)
Gear coupling for large shaft separation (Figure 6.23)
Flangeless gear coupling (Figure 6.24)

b. Special-Purpose High-Speed Gear Couplings These couplings are usually available in sealed lubrication and continuous lubrication types in four basic styles (see Figure 6.25) Most of these couplings conform to the requirements of API 671. They include:

The spacer coupling for flow-through continuous lube (Figure 6.26A)

The spacer coupling for packed lube (Figure 6.26B)
The spacer coupling for dual-direction continuous lube (Figure 6.26C)
The spacer coupling for dual-direction continuous lube (Figure 6.26D)
The spool-type coupling for dual-direction continuous lube (Figure 6.26E)
The spacer coupling for dual-direction continuous lube with reduced overhung moment on connected shafts (Figure 6.26F)
The spool-type coupling for packed lube (Figure 6.26G)

These couplings are usually made from alloy steels and operate at speeds usually in excess of normal motor speeds.

Two typical applications are in gas turbines and in centrifugal compressors.

Figure 6.27 shows a gas turbine. Gear couplings are used between the turbine and whatever it drives: gear box, compressor, generator, or pump. Figure 6.28A shows a typical load coupling, and Figure 6.28B a typical accessory coupling. These couplings are required to take high loads at startup and must accommodate large axial movements (1/2 to 1 in.) while having very low axial reactionary loads. This particular turbine has been around for over 30 years and over this time frame the horsepower of the turbine has almost doubled. The gear coupling for this application has been improved in design without changing its size. Higher-strength material is used. The geometry of the crowned tooth has improved. And the tooth spacing and profile has been improved by lapping sets together.

The other typical application of using high-speed gear couplings is in the centrifugal compressor. (Figure 6.29).

c. High-Torque/High-Angle/Low-Speed Gear Couplings These gear couplings are very common on steel mill drives (see Figure 6.30). There are two basic types: standard lines (flange types) similar to those shown in Figure 6.31 and designed styles (Figure 6.32), which are used for primary mill roll drives. The flange-type flexible spindle (Figure 6.30) is similar to a tandem arrangement using flange-type couplings except that the gearing is cut to accommodate higher misalignment. The gear teeth are heat-treated to provide higher torque ratings, and special seals, molded high-angle lip-type seals or specially designed rising ring types, are used. This spindle is for medium-duty, medium-torque applications with relatively high misalignment capacities required. It is used in applications where equipment is not subjected to frequent disconnecting of drive components—primarily auxiliary equipment such as floating shafts for pinch rolls, tension bridles, continuous casting equipment, plastics and rubber calendars, rotary side guides, paper mill equipment, and auxiliary equipment applied on electrolytic cleaning, pickling, and galvanizing lines. The custom-designed gear spindle (Figure 6.32) is used on heavy-duty high-torque applications requiring rugged strength, such as in metal rolling mill main drives and similar heavy equipment. Figure 6.33 shows some typical dimensions for this type of spindle. Note that these dimensions vary for various gear spindle vendors and the type of mill the spindles are used on. These spindles are not normally interchangeable. To assure interchangeability when changing from one vendor to another, parts can be measured and "reverse" engineering can be done to assure interchangeability. Mill-type flexible spindles deliver maximum torque capacity at relatively high operating angles and under severe shock load conditions on all types of equipment for the metals, rubber, plastics, paper, automotive, and chemical industries.

Spindle gear couplings show the following variations and features:

1. *Splined hubs* (Figure 6.34A): Male gear hubs are connected to the intermediate shafts through snug-fitting splines employing a side bearing fit held concentric with the spindle shaft by pilots on both ends of the spline. Thus the hub is held in a fixed position, able to resist radial load. Because of these pilot diameters, the spline is used only to transmit torque. Relative movement between the male and female spline is minimized, ensuring that fretting and spline wear cannot take place. Also, the spline requires less radial space than do keys, thus permitting the use of large shafts. Whenever possible, spindles are manufactured with hubs retained, with no internal bolts externally. This is done by using a large nut threaded on the end of the shaft or a split retainer that fits in a groove in the shaft and is through-bolted into the back of the hub. This eliminates all internal bolting and the possibility of broken capscrews working into the gear teeth, causing premature failure.
2. *Insert gear rings* (Figure 6.34B): This component is con-

nected to the respective sleeve or adapter, usually through a snug-fitting spline employing a side-bearing fit. The gear ring is usually held concentric with the adapter by tight-fitting splines or by using a pilot, usually at both ends of the splined engagement.

3. *Spring-loaded thrust button* (Figure 6.34C): Many spindles are designed to incorporate a spring-loaded thrust button in the roll end coupling that holds the roll end coupling in line with the spindle shaft during roll change. This button is actuated by a spring force when the roll neck is removed from the coupling. The force of the spring drives the sleeve assembly forward until the sleeve assembly locks against the hub element for a full 360° contact, thrusting it into a horizontal position, ready to receive a new roll. Most high-angle spindles (generally over 1°) are designed so that the contact point of the thrust surface is located on the centerline of the gear mesh. This provides ball-and-socket action rather than the sliding motion that occurs if the contact point is located back on the thrust plate.

4. *Splined replaceable sleeve* (Figure 6.34D): Many spindles are designed with a splined replaceable sleeve on one or both ends. Basically, this is an adapter that fits onto the roll or the pinion shaft and intermediate sleeve, is splined to the adapter, and has an internal gear that meshes with the hub. This type of design is desirable when a quick-disconnect feature is required.

5. *Seals:* Seal for high angularity are very special in design. High-angle lip seals (Figure 6.34E) are common. The seal is an integral one-piece molded lip seal, usually designed specifically for a particular flexible spindle application. Rising ring seals (Figure 6.34F) are also very common; the "rising" ring can be made of bronze, plastic, etc. Both of these seals can be made in split form for ease of maintenance, but some sealing advantages are sacrificed by doing this. Generally, split seals leak and therefore require more frequent lubrication.

6. *Roll end bores* (Figure 6.34F): There is much controversy as to whether one wants the roll to wear first or the bore of the spindle: Should one make the bores (flattened portion) soft or hard? Should one use key inserts or harden the bore? Many spindles are furnished with either developed (shaped) bores or replaceable wear keys that can be hardened or that use standard commercial steels, depending on the application. Roll end bores of the developed bore configuration are generally used with some type of surface heat treatment. The heat treatment possi-

bilities include nitriding or carburizing. Nitrided bores require no correction after nitriding and are generally 0.035 in. maximum in case depth but can be as hard as 68 Rc (Rockwell hardness—C scale). They provide excellent wear characteristics. Carburized bores usually require correction after carburization. Bores can be carburized to a hardness of 62 Rc and depths of 0.200 in. But after grinding the hardness may be 58 Rc with case depths in some areas as low as 0.090 in. Both types show similar lifetimes in the same application. This is attributed to the better wear characteristic of the nitrided bore and the uniform hardness of the carburized case depth. Therefore it is established that 0.090 in. is the maximum acceptable bore wear a nitrided bore will take two or three times before it wears 0.030 in., then at a much faster rate beyond the case. The carburized bore wears at the same rate throughout the 0.090 in. In general, at the 0.090 in. wear limit the time would be about the same for both nitrided and carburized bores. The lifetime also depends on the materials used. For example, Nitralloy 135 bores have lasted longer than 8620 carburized and ground bores. And 3310 carburized and ground bores have lasted longer in some applications than Nitralloy 135 bores.

What type of roll bore is used usually comes down to experience or preference. Bores with key inserts are most commonly used. These are usually bar mills, section mills, and strip mills. Rough stands and reversing mills commonly use shaped bores.

d. High-Angle/High-Speed Gear Couplings This type of gear coupling is typically used on helicopters and on traction drives for subway cars. Helicopter couplings include accessory drive couplings, main drive couplings, and tail rotors. Figure 6.35 shows drawings and schematics of some of these; Figure 6.36A, a main drive helicopter coupling; Figure 6.36B, the helicopter that uses this coupling; and Figure 6.37, a traction coupling design and application. Figure 6.35A is a simplified drawing of the drive train of a typical turbine-powered helicopter utilizing three types of high-performance gear couplings. These types are listed and described briefly below.

1. *Main drive* (Figure 6.35B): Connects the engine to the rotor gearbox. Fins on the sleeve aid in heat dissipation and stiffening (rated at 2020 hp at 6360 rpm with 2° of angular misalignment per gear mesh).
2. *Accessory drive* (Figure 6.35C): Connects the turbine-engine-

driven main gearbox to the accessory gearbox (rated at 130 hp at 6600 rpm with 2° angular misalignment per gear mesh).

3. *Tail rotor drive* (Figure 6.35D): Rated at 185 hp at 3130 rpm with 2° angular misalignment per gear mesh. A series of these single engagement couplings are used in the tail rotor drive system to compensate for air frame deflection.

e. Gear Couplings for Special Applications Gear couplings have been designed to do many things and accommodate many situations. Some of these are:

1. Brake drum and brake disk incorporated into gear coupling (Figure 6.38)
2. Insulated gear coupling: driver and driven insulated from stray electric currents (Figure 6.39)
3. Limited-end-float-position motor rotors that do not have thrust bearings (Figure 6.40)
4. Shear pin coupling for overload protection (Figure 6.41)
5. Disconnects (Figure 6.42)
    a. Low-speed disconnect (Figure 6.42A): provides manual or automatic disengagement of driver and driven, usually done at 0 rpm
    b. High-speed disconnect (Figure 6.42B): provides automatic disengagement of driver and driven, usually done at 0 rpm or at very low speeds

E. Materials and Heat Treatment for Gear Couplings

1. Materials

The various materials used for gear couplings include the following:

a. *As rolled or normalized, AISI 1035 to 1050:* This material is used on standard general-purpose gear couplings. Couplings using this material are generally lightly torqued and cut for less than 1½° up to 14 PD; over 14 PD they are usually cut for less than ¾° misalignment (Figure 6.43A).

b. *Flank-hardened, AISI to 1035 to 1050 (usually induction hardened), 50 Rc:* This material is used to improve wear. Couplings using flank-hardened teeth are usually rated for light torque, low speed, and high angle (up to 6°) (Figure 6.43B).

c. *Fully contoured induction hardened, AISI 1035 to 1050, 50 Rc:* This material is used for steel mill applications. It not only improves wear but increases the bending strength of teeth. This process is limited to medium-crowned teeth (usually under 4°) and tooth size between 5 and 2.5 DP. Couplings using this material are usually rated for moderate torque, low speed, and high angle (Figure 6.43C).

d. *Heat-treated AISI 4100 and 4300 steels, core 28 Rc (usually 4140 and 4340):* This material usually increases capacity slightly but does not affect wearability. Couplings of this material are usually applied to medium-torque, medium-speed, low-angle applications (Figure 6.43D).

e. *Flank-hardened (induction hardened) AISI 4100 and 4300 steels, surface 50 Rc, core 28 Rc (usually AISI 4140 or 4340):* This material is used on steel mill applications to improve wear. Couplings using flank-hardened teeth of this material are usually applied to moderate-torque, low-speed, high-angle applications (Figure 6.43E).

f. *Fully contour (induction-hardened) AISI 4100 and 4300 steels, surface Rc, core 28 Rc (usually AISI 4140 or 4340):* This material, used for steel mill applications, not only improves wear but increases the bending strength of teeth. This process is limited to medium-crowned teeth (under 4°) and tooth sizes between 5 and 2.5 DP. Couplings using this material are usually applied to high-torque, low-speed, high-angle applications (Figure 6.43F).

g. *Nitrided AISI 4100 and 4300, surface 54 to 60 Rc, core 35 Rc (usually 4140 or 4340); case 10% tooth thickness and 0.035 in. maximum depth, maximum tooth size 5 DP:* This material gives a coupling increased strength in the root. Couplings of this material are usually used for small steel mill couplings with high torques, medium speed, and high angles (Figure 6.43G).

h. *Nitrided AISI 4100 and 4300, surface 54 to 60 Rc, core 28 to 32 Rc (usually AISI 4140); case depth 0.010 to 0.035 in. depending on tooth size:* This material takes heat treating well and produces little distortion. Couplings of this material are used in a wide range of applications: light to moderate torque, low to high speeds, and low to high angles (Figure 6.43H).

- i. *Nitralloy (135 or N), surface 60 Rc, core 32 to 45 Rc; case depth 0.010 to 0.035 in. depending on tooth size:* This material provides excellent wear resistance and produces very little distortion. Couplings of this material are used in high-torque, high-angle, low-speed applications such as gear spindles or in high-angle, high-speed ones such as helicopter and traction couplings (Figure 6.43I).
- j. *Carburized AISI 8620, 4320, 3310, and others, surface 58 Rc, core 32 to 38 Rc; case depth 0.062 to 0.250 in. depending on tooth size:* These materials are used in many steel mill applications. Couplings made of these materials usually require grinding or lapping after carburizing to correct for tooth distortion caused by the carburizing process (Figure 6.43J).

Figure 6.44 shows the relative strength of various materials and heat treatments compared to as-rolled (1035 to 1050) steel. Figure 6.45 shows the relative wear characteristics of various materials and heat treatment.

2. Heat Treatment

Basically, heat treatment of steel consists of raising the material some specified temperature. The process is performed to change certain characteristics of steel to make them more suitable for a particular kind of service. Some of the reasons for heat treating are:

1. To soften a part so that it can be machined more easily
2. To relieve internal stresses so that a part will maintain its dimensional stability (i.e., not warp or fail prematurely due to locked-in stresses)
3. To refine the grain structure so that the part will be less apt to fracture abruptly (or to toughen)
4. To thoroughly harden a part so that it will be stronger
5. To case harden a part so that it will be more wear resistant

3. Some Heat Treatment Processes

Annealing: Generally refers to the heating and controlled cooling of a material to remove stresses, make it softer, refine its structure, or change its ductility, toughness, or other properties.

Carburizing: Adding carbon to iron-base alloys by absorption through heating the metal at a temperature below its melting point in contact with carbonaceous materials. Such treatment followed by appropriate quenching hardens the surface of the metal. This is the oldest method of quenching.

Flame hardening: In this method of hardening, the surface layer of a medium- or high-carbon steel is heated by a high-temperature torch and then quenched.

Induction hardening: A hardening process in which the part is heated above the transformation range by electrical induction.

Nitriding: Adding nitrogen to solid iron-base alloys by heating at a temperature below the critical temperature in contact with ammonia or other nitrogenous material.

Hardening(as applied to the heat treatment of steel): Heating and quenching to produce increased hardness and increased strength, respectively.

Normalizing: Heating to about 100°F above the critical temperature and cooling to room temperature in still air. Provision is often made in normalizing for controlled cooling at a slower rate, but when the cooling is prolonged, the term used is *annealing.*

Stress relieving: Reducing residual stresses in a metal by heating to a suitable temperature for a certain period. This method relieves stresses caused by casting, quenching, normalizing, machining, cold working, or welding.

Tempering: Reheating after hardening to a temperature below the critical temperature and then cooling.

4. The Importance of Heat Treatment

The best heat treatment for a coupling gear tooth gives the correct combination of core hardness versus case depth and hardness. If the core is too soft, the wear and load-carrying capabilities of a coupling become limited because the core will cause the coupling to fail in subsurface shear instead of compression

Normalizing and/or annealing: Contoured or forged-bar materials that are subsequently surface hardened to be used in couplings are usually normalized or annealed. This assures a refined grain structure that helps throughout the manufacturing cycle. The refined grain structure helps produce a good finish and minimizes localized cold working, which can produce stress risers that can cause distortion in parts after final heat treating or machining and premature failure from cracks produced by the stress risers.

Stress relieving: This process is used for material that is heat-treated to a high core hardness and is usually used in high-angle, high-speed couplings. Parts are stress-relieved after rough machining and/or before case hardening. This reduces Internally locked-in stresses caused by core hardening and machining. Stress relieving helps prevent distortion or cracking after final heat treating and machining.

Hardening and tempering: Steel is usually hardened and tempered to obtain the desired level of strength. This process is used to increase the strength capacity not only of gearing but also of such torque transmission parts as spacers, hubs, and rigid coupling. With the increase in core strength there is a reduction In ductility or impact resistance. This is one reason why parts are not hardened to maximum hardness throughout. Depending on the material, typical heat-treated-material core hardness range from 28 to 38 Rc. Beyond 38 Rc, not only does ductility decrease but normal machining becomes almost impossible.

Case hardening of gear teeth: Why surface-harden gear teeth? The main purpose is to impart a more highly wear resistant surface. Increased wearability can be obtained by hardening the gear flank surface by one of the following methods: flame or induction hardening, fully contoured induction hardening, nitriding, or carburizing.

5. Benefits of Surface Hardening

An additional benefit can be derived from the surface hardening process if a uniform, adequate increase In tensile strength can be obtained. The hardened surface and the ductile core must be properly balanced to provide a high-strength ductile tooth for bending.

A rectangular cantilever beam subject to bending will result in the stress distribution shown in Figure 6.46A. The greatest stress occurs at the outer fiber and decreases linearly to zero at the neutral axis or center. The maximum stress without permanent deformation is the yield stress of the material. Figure 6.46B illustrates the same cantilever surface hardened. The stress distribution is similar, but the maximum stress without permanent deformation is not the yield of the core material but the yield of the surface and the ductile core. A gear tooth is like a cantilever beam. A properly balanced tooth is one in which the extreme fiber stress reaches the yield strength of the hardened material when the stress at the juncture of the core and the surface reaches the yield strength of the core. It can be shown algebraically that the proper case-hardened depth should be approximately one-tenth to one-fifth of the tooth thickness for surface-hardened alloy gear teeth. For all case hardening listed below, not only is wear resistance increased but tensile strength is also increased, by at least 50% (see Figure 6.45).

1. Fully contoured (bending load) induction hardened 5 to 2.5 DP teeth cut for 4° or less of misalignment
2. Nitriding smaller than 5 DP
3. Carburizing all

6. Surface Hardening

a. Hardening of Medium-Carbon Steels. How is the gear tooth hardened? First, the part is heated to above the transformation temperature. This temperature varies with different grades and types of steels, but it is the temperature at which the structure of the steel changes and carbon goes into solution. This simply means that the carbon atom is freed to move about in the steel. For nearly all steels processed, this temperature is about 1500°F. The next step after heating is quenching or very rapid cooling. In this process excess carbon atoms are trapped and cannot return to their original locations. If the quench is ideally rapid and the part is cooled to a low temperature, there will exist a condition of 100% martensite. This is the strongest of all structures, but it is very difficult to obtain and not at all desirable for couplings because it is very brittle and subject to impact failures.

b. Flame Hardening. Flame hardening is a surface-hardening process in which a steel is rapidly heated above its transformation temperature by a high-temperature flame and then cooled at a rate to produce the hardness desired. Flame-hardened gear teeth can be made to 50 to 56 Rc. Because the heat is localized, there is relatively low distortion, but there is a great difficulty in producing a repeatable concentration of heat. There can not only be hardness variation from tooth to tooth but also hardness variation within a single tooth. This process does add wear resistance, but the overall strength of the base material is lowered.

c. Induction Hardening. Induction heating takes place when an electrically conducting object (not necessarily magnetic steel) is placed as a varying magnetic field. Induction heating is due to hysteresis and eddy current losses; and although it has been in wide use for more than 25 years, this heating effect is not completely understood. Probably the primary reasons for using this type of heating are that it is fast and very easy to apply to a particular area. Since a gear tooth should be case-hardened without affecting its resilient core, it is necessary to apply the heat very rapidly so that the cooler center of the gear does not rob the heat. Because the heat pattern must be controlled very closely, induction heating is ideal for this process.

Conventional flank hardening. Hardening the flanks of a gear tooth by induction hardening, shown in Figure 6.47A, can provide a hardened surface for increased life. The lack of hardness in the root of the tooth usually not only does not improve the strength of the tooth, but can reduce its strength. To understand this, you have to realize that hardening creates stresses. The hardening process is performed by heating a steel part to an elevated temperature, usually around 1500°F and then rapidly quenching the part. The heated and quenched portion actually Increases In volume, setting up compressive stresses internally. It becomes a very dense, hard structure, and its strength increases In proportion to its Increase in hardness. In conventional processes, there could be sharp stress discontinuities if this tooth is highly loaded at the root area. These discontinuities could be starting points for stress cracks.

Fully contoured hardening. Fully contoured induction hardening is an answer to the need for gear tooth wearability and strength. Figure 6.47B Illustrates the fully contoured pattern. Observe that the hardened case is contoured through the root and a transition between the hard case and the soft core occurs at the unstressed tooth tip. The process is carried out by core hardening the rough turned forging to the upper machineability limit, and then after cutting the gear teeth, hardening the coupling one tooth space at a time. The absence of large quantities of heat results in a gear essentially free of overall distortion. The depth of the case can be controlled to permit long wear life and a strong gear tooth in bending for impact loads.

d. Nitriding. Nitriding is a process used for case hardening of alloy steel parts usually of special composition, by heating them in an atmosphere of ammonia gas and disassociated ammonia mixed in suitable proportions. The process is carried out below the transformation range for steel and quenching is not involved. Nitriding is particularly desirable because of the following features:

1. Exceptionally high surface hardness
2. Very high wear resistance
3. Low tendency to seizing and galling
4. Minimum warpage or distortion
5. High resistance to fatigue
6. Improved corrosion resistance

Nitriding a gear coupling is done selectively. That is, all areas except the gear tooth are blocked off with paint or plating during nitriding. This procedure minimizes part distortion and allows conventional machining in critical areas such as bolt holes and pilots to be done after nitriding.

In nitriding the depth and hardness of the case is a function of two things: the nitriding process time and, in some alloys, the hardness of the core. The maximum case depth is approximately 0.035 In. for material with a core hardness above 36 Rc. With the use of special nitriding steels such as Nitralloy, surface hardnesses as high as 65 and 70 Rc are attainable. If we apply the rule for depth of case versus tooth size, case depth should be approximately one-tenth of the tooth thickness. We can see that with a maximum surface depth of 0.035 in., we will not obtain any increase in bending capacity at the root of the tooth for DP values larger than 5 DP. Normally, nitriding is very advantageous to increase the wear resistance of high-speed, high-angle couplings but it can supply good fatigue resistance at tooth roots for small DPs (smaller than 5 DP). For high-angle, high-speed applications, coupling materials like

Nitralloy N are used. This material precipitates hardness during the nitriding process (the core increase from 25–30 to 38–45 Rc. This produces a very strong core that supports the case under high-impact loads.

e. Carburizing. Carburizing is the ultimate means of hardening a coupling tooth as it can produce the balance needed for wear resistance and strength. Its use results in a very hard, deep case (as high as 65 Rc) and a tough ductile core. The uniform hardness pattern at the flanks extends through the root of the tooth. The depth of hardness can be controlled by the amount of time in the furnace. Wear has been found to be a function of hardness and carbon content. Figure 6.48 shows the wear resistance of several materials as a function of hardness. The major problem with carburizing is that the entire part must be placed in a furnace at a high temperature. A fully machined gear tooth is subjected to approximately 1700°F for many hours and is then quenched in oil. As one would expect, a great deal of distortion results. Gear coupling designs require uniform tooth loading to survive the rigors involved in many applications. The distorted carburized gear cannot produce uniform tooth loading. The few teeth carrying the load must "wear in" to allow other gear teeth to share in the load. This early wear can result in reduced gear life. Usually some correction for distortion is made after carburizing. Figure 6.49 shows blue contact check on three gear spindles. Figure 6.49A shows the pattern of a set of gears as carburized 20 to 40% gear tooth contact, Figure 6.49B shows the pattern of a set of gears that have been lapped after carburizing, 60 to 70% gear tooth contact. And Figure 6.49C shows the pattern of a set of gears that were ground (to AGMA 10 quality) after carburizing, 90 to 100% gear tooth contact. Figure 6.49D shows the wear pattern of a carburized and ground spindle in a hot mill finishing stand after one year of operation.

F. How to Evaluate Gear Couplings

1. Comparison Method

One question often asked is how to evaluate or compare a gear coupling. It is very difficult to give an answer because no two gear couplings are exactly alike. What I have attempted to supply is a procedure that can be used to compare couplings. Most gear couplings fail because of tooth distress, breakdown of lubricant, compressive stress, and the motion due to misalignment. If an overload occurs, the gear coupling will fail at its weakest link:

1. Tooth shear or bending
2. Bolts
3. Shaft or spacer
4. Flanges
5. Hubs
6. Key or keyways

Items 2 through 6 are covered in Chapter 3, Section II. Failures of gear teeth in shear or bending are uncommon. The stress equations vary due to the tooth stress used. The stresses are a function of:

Bending stress:

$$S_b = K_b \frac{T}{(\mathrm{PD})^2 F} = \mathrm{psi} \tag{6.1}$$

Shear stress:

$$S_s = K_s \frac{T}{(\mathrm{PD})^2 F} = \mathrm{psi} \tag{6.2}$$

where

T = torque (in.-lb)
PD = pitch diameter (in.)
F = face width (in.)
K_s and K_b are a function of:

1. The amount of crown
2. The number of teeth in contact
3. Misalignment
4. Other geometric variables

For high-angle couplings (generally operating over 1°) the coupling rating is usually limited by the coupling tooth bending strength. Therefore, one should ask for the couplings endurance limit and yield limit. Because some cases the weak link may be a part other than the gearing, (it may be the shaft, spacer, bolts, hub, etc.), one should determine what the limits are for the weakest link.

2. Criteria for Comparison

The following criteria give a relative comparison of the wear life of a gear coupling. First obtain the following information:

T = raw motor torque per spindle (in.-lb)
PD = diameter of pitch circle of gear tooth (in.)
h = active tooth height (in.)
D = diameter of curvature of tooth face (in.)
DP = diameter pitch of gear tooth
N = maximum speed of spindle
α = full-load misalignment angle (deg)
FW = gear tooth face width (in.)
R = PD/2 (in.)
n = number of teeth in coupling

For crowned teeth (based on the compressive stress equation in MIL-C-23233):

$$S_c = 2290\sqrt{\frac{T}{[\mathrm{PD}/2] \times (D/2) \times c \times h \times (\mathrm{DP} \times \mathrm{PD})}} = \mathrm{psi} \qquad (6.3)$$

where c represents the percentage of teeth in contact (see Figure 6.50).

If tooth curvature Is expressed in amount of crown:

K = amount of crown = (in.) (see Figure 6.51)

$$D = \frac{(\mathrm{FW}/2)^2}{K} = \mathrm{in.} \qquad (6.4)$$

$$h = \frac{1.8}{\mathrm{DP}} = (\mathrm{in.})$$

$$S_c = 3600\sqrt{\frac{T}{(\mathrm{PD})^2 \times D \times c}} = \mathrm{psi} \qquad (6.5)$$

For straight-tooth couplings:

$$S_c' = \frac{T}{RnA} = \mathrm{psi} \qquad (6.6)$$

$$A = \frac{1.8 \times \mathrm{FW}}{\mathrm{DP}} = \mathrm{in.}^2 \qquad (6.7)$$

For the maximum sliding velocity V of the gear coupling:

$$V = \pi \times \mathrm{PD} \times \sin \alpha \times \frac{N}{60} = \mathrm{ips} \tag{6.8}$$

S_cV criteria can be used to compare equivalent couplings and applications:

$$S_cV = S_c \times V = \mathrm{psi} - \mathrm{ips} \tag{6.9}$$

Limits on S_cV depend on the type of lubrication and the material used for a particular coupling. Some typical values based on testing and field experience are given in Figure 6.52. It should be remembered that the allowable values change with material, tooth geometry, tooth finish, and type of lubrication.

3. Torque Capacity

The torque capacity T can be approximated by using equation (6.10) and Figures 6.53, 6.54, and 6.55.

$$T = T_B \times S \times M \tag{6.10}$$

where

T_B = base torque (from Figure 6.53)
M = misalignment factor (from Figure 6.54)
S = material factor (from Figure 6.55)

This equation and the factors in Figures 6.54 and 6.55 are empirical, based on years of experience.

G. How Gear Couplings Affect the System

The most significant effect that a gear coupling has not only on itself but also on the system comes from the moments and forces generated when it slides and/or misaligns.

1. Axial Force

The formula for the axial force reacted back onto the thrust bearings is

$$F = \frac{T\mu}{R} \tag{6.11}$$

where

T = torque (in.-lb)
μ = coefficient of friction
R = pitch radius of gear (in.)

Test data available from various coupling manufacturers show that typical values for coefficient of friction are:

Sealed lubricated gear couplings:

$\mu = 0.05$

Continuous lubricated gear couplings:

$\mu = 0.075$

For equipment design purposes it should be safe to multiply the foregoing values by 2. Values higher than these can be experienced, but if they are seen for any period of time, the coupling is no longer flexible and more than likely will fail or cause a failure of some portion of the equipment. Experience indicates that for values of μ above 0.15, a failure of some type could occur. In fact, if a coupling is locked (mechanically) from sludge or wear, the forces experienced could be seven to eight times those normally expected.

There is much discussion of how large a value to use when designing a system or, in fact, when designing the coupling. Designing as if the coupling were locked would completely defeat the purpose of a flexible coupling. It would be like trying to design rotating equipment to run with no lubrication in the bearings. If couplings are lubricated and maintained properly, and operated within the loads and misalignment values used to select them, the coefficient of friction should never exceed 0.15. It is up to the system designer to decide how safe is safe. This must be decided on the basis of what would happen if a failure were to occur.

2. Bending Moment

The moments produced in a gear coupling can load equipment shafts and bearings and change the operating characteristics of equipment. First, let us define a coordinate system for a gear coupling. In Figure

6.56, the line X' defines a cut through the plane of misalignment. Moments that tend to rotate the coupling around the Z–Z axis will be designated M. Moments that tend to rotate the coupling around the Y–Y axis will be designated M_S.

Gear tooth sweep:

$$\sin \alpha = \frac{x}{R_c} \tag{6.12}$$

$$R_c = \frac{x}{\sin \alpha} \tag{6.13}$$

$$D_c = \frac{2x}{\sin \alpha} \tag{6.14}$$

$$2x = D_c x \sin \alpha \tag{6.15}$$

where x equals the distance from the centerline of the gear tooth to the point along the flank of the gear contact at α° misalignment (see Figure 6.57)

The average radius for a misaligned coupling (see Figure 6.58):

$$R' = \sqrt{R^2 - \left(\frac{R}{3} \sin \frac{A}{2}\right)^2} \tag{6.16}$$

where

R = pitch radius of gear (in.)
A = angle for gear tooth contact range (deg)

There are three basic moments in a gear coupling:

1. The moment generated from transmission torque and angle (rotated around the Z–Z axis)
2. The moment generated from frictional loading (rotated around the Z–Z axis)
3. The moment generated from displacement of load from its condition rotating around the Y–Y axis

The forces on the coupling tooth are (Figure 6.59):

$$\Sigma F_{X'} = +P \cos \alpha - \mu N - W \sin \alpha = 0 \tag{6.17}$$

$$\Sigma F_{Z'} = N - P \sin \alpha - W \sin \alpha = 0 \tag{6.18}$$

$$N = P \sin \alpha + W \cos \alpha \tag{6.19}$$

Eliminating N from equations (6.16) and (6.17), we have

$$P \cos \alpha - \mu(P \sin \alpha + W \cos \alpha) - W \sin \alpha = 0 \tag{6.20}$$

$$P \cos \alpha - \mu P \sin \alpha = +W \cos \alpha - \mu W \sin \alpha \tag{6.21}$$

$$P = \frac{W(\sin \alpha + \mu \cos \alpha)}{\cos \alpha - \mu \sin \alpha} \tag{6.22}$$

Moments causing rotation around the Z–Z axis

$$M_P = P \times 2R' \tag{6.23}$$

$$M_P = \frac{2W(\sin \alpha + \mu \cos \alpha)}{\cos \alpha - \mu \sin \alpha} \sqrt{R^2 - \left(\frac{R}{3} \sin \frac{A}{2}\right)^2} \tag{6.24}$$

$$M_P = \frac{M_t (\sin \alpha + \mu \cos \alpha \sqrt{R^2 - [(R/3) \sin (A/2)]^2}}{R (\cos \alpha - \mu \sin \alpha)} \tag{6.25}$$

Moments causing rotation around the Y–Y axis are:

$$M_S = M_f \alpha + M_G = 0 + M_G = W(2x) \qquad \text{where} \quad W = \frac{M_t}{2R} \tag{6.26}$$

$$M_S = \frac{M_t x}{R} \tag{6.27}$$

The resultant moment is the vectorial combination of two moments:

$$M_2 = \sqrt{(M_S)^2 + (M_P)^2} \tag{6.28}$$

$$\text{at approximately } \theta = \tan^{-1}\left(\frac{2R}{D_c}\right) \text{ from } Y \text{ in the } ZY \text{ plane.} \tag{6.29}$$

The final combined moment equation is:

$$M_2 = \left[\frac{M_t(x)}{R}\right]^2 + \left[\frac{M_t (\sin \alpha + \mu \cos \alpha)}{R(\cos \alpha - \mu \sin \alpha)} \sqrt{R^2 - \left(\frac{R}{3} \sin\left(\frac{A}{2}\right)\right)^2}\right]^2 \tag{6.30}$$

or

$$M_2 = \sqrt{\left[\frac{M_t(x)}{R}\right]^2 + \left[\frac{M_t\,(\sin\alpha + \mu\cos\alpha)}{R\,(\cos\alpha - \mu\sin\alpha)}\,(\mathrm{R}')\right]^2}$$

3. Reactionary Forces on Equipment Bearings

Since maximum moment occurs at $\theta = \tan^{-1}(2R/D_c)$ from Y in the ZY plane, maximum bearing loads will be perpendicular. Pictorially this is difficult to show, so we will show loads as if they are parallel to the Z axis. This does not affect the magnitude of the load, only the point of application, and is not very important. We will solve bearing reactions In terms of maximum moment, as if the maximum moment were tending to rotate the coupling around the Y–Y axis.

Considering Figure 6.60 (plan view), with torque and rotation directions shown as indicated and an α° plane of misalignment as given, it may be observed that the force pattern nets a clockwise moment transferral (M) to the shaft in an amount. The bearing nearest the transferral moment would then have a high-pressure condition, B, in the Z plane, and loading with the front bearing down and the back bearing up. This assumes the direction of rotation to be over the top of the shaft. Refer to Figure 6.60 for further details.

Taking the ΣM values about A and setting them equal to zero, we have

ΣM about $A = B'L - M = 0$

$B' = \dfrac{M}{L}$ resisting force required of bearing

$b' = A'$ equal and opposite in direction

The resulting force pattern for A and B would then be the opposite of C and D, reading from left to right. Figure 6.60 indicates the force pattern that results for the direction of rotation shown. When the direction of shaft rotation is reversed with the same α° misalignment condition, the forces reverse themselves as indicated.

H. Failure Mode for Gear Couplings

The most common type of failure for gear couplings is tooth distress (Figure 6.61 shows typical good wear patterns). Common types of failures are:

1. Failure due to lubricant breakdown (Figure 6.62).
    a. Loose.
    b. Lubricant washed out.
    c. Lubricant deterioration.
    d. Improper lubricant used.
2. Failure due to improper contact (Figure 6.63). In this case the matched lapped set was mixed. The resulting failure was not tooth distress itself but increased moment, which broke the coupling spacers.
3. Failure due to lack of lubrication. Lube nozzles were misdirected and the coupling was lubricated only by an oil mist (Figure 6.64).
4. Failure due to overmisalignment, in which the ends of the teeth are broken or cracked (Figures 6.65 and 6.66).
5. Failure due to overmisalignment during roll change causing cracks that precipitate to total failure during normal operation (Figure 6.67).
6. Failure due to operating at torsional critical levels at which the lubricant breaks down and cold welding takes place (Figure 6.68).
7. Worm tracking (Figure 6.69). Cold flow or welding usually occurs on continuously lubed couplings. This occurs near the end of the teeth when misalignment approaches the design limit of the crown. At this point the lubrication film breaks down and causes metal-to-metal contact. Sometimes cold flow distress that takes place anywhere on a tooth is described as worm tracking. Cold flow occurs when the lubricating film breaks down. This can result from high localized tooth loading or from lubricant deterioration.
8. Occasionally an overload can cause a gear tooth to shear off (see Figure 6.70).
9. Sludge buildup can cause failure of coupling components other than teeth or equipment components such as shafts or bearings (see Figure 6.71). Once a gear coupling locks up because of sludge, anything can break (see Figure 6.72). Sludge also collects corrosive residue, which can corrode coupling parts and act as a source of crack initiations for fatigue-propagated failure of a part (see Figure 6.73).

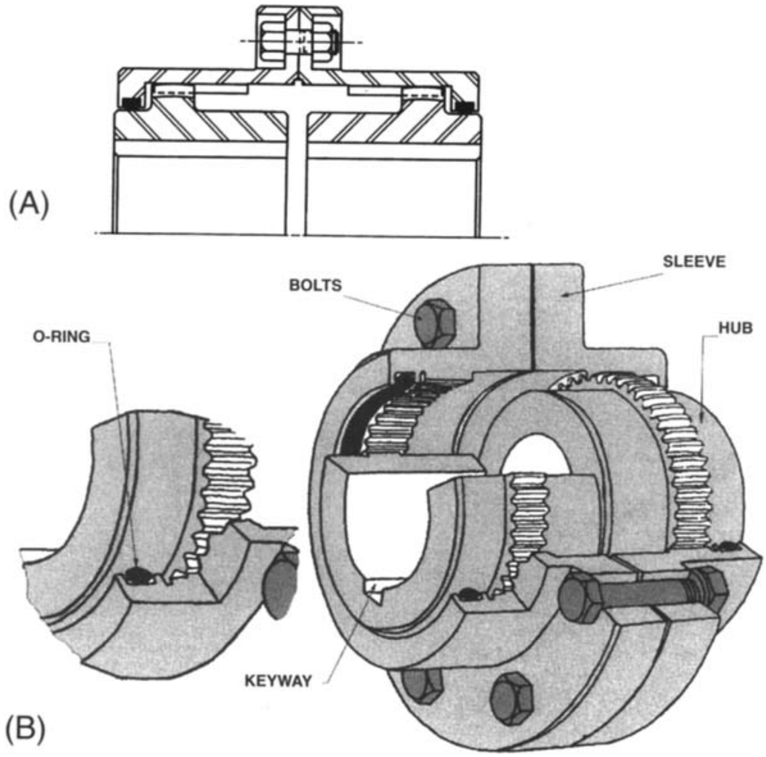


Figure 6.1 (A) Typical gear coupling. (B) Typical O-Ring Gear Coupling.

10. Lockup can also be caused by tooth wear. The hub tooth embeds (wears) into the internal gear tooth, which causes mechanical lockup (Figure 6.74).
11. Other type of gear coupling failures include:
    a. Fatigue failures through gear teeth (Figure 6.75).
    b. Key shear or roll (Figure 6.76A).
    c. Damage to key, shaft, and hubs caused by improper fit (Figure 6.76B).
    d. Hub bursting, usually caused by overload on couplings that are used with maximum bore and keyway sizes. The hub may be the weak link. Hubs split over corners, and in some cases this wedging can cause a sleeve to split (Figure 6.77).

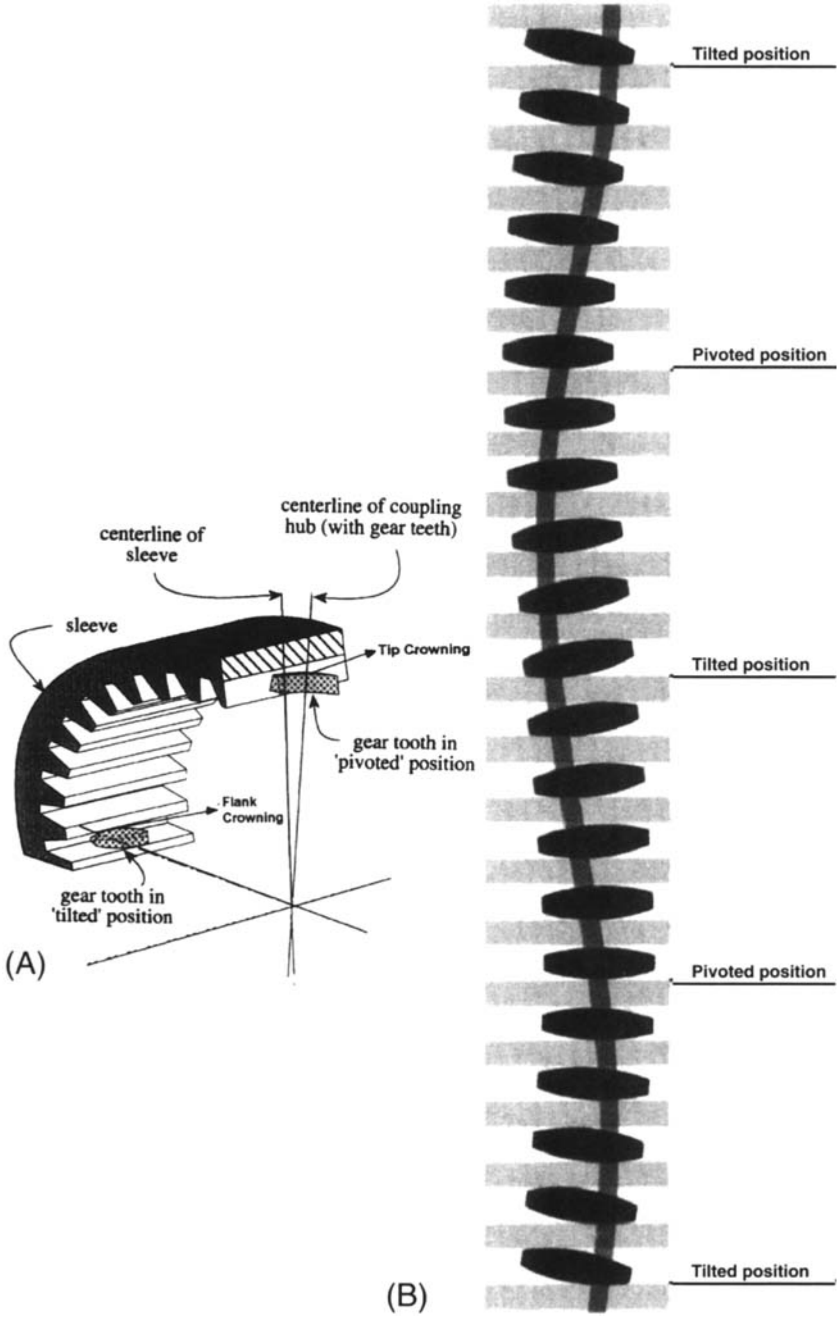


FIGURE 6.2 (A) Contact of misaligned gear coupling, no contact in pivot position. (B) Tooth rollout showing tooth contact of a misaligned gear coupling.

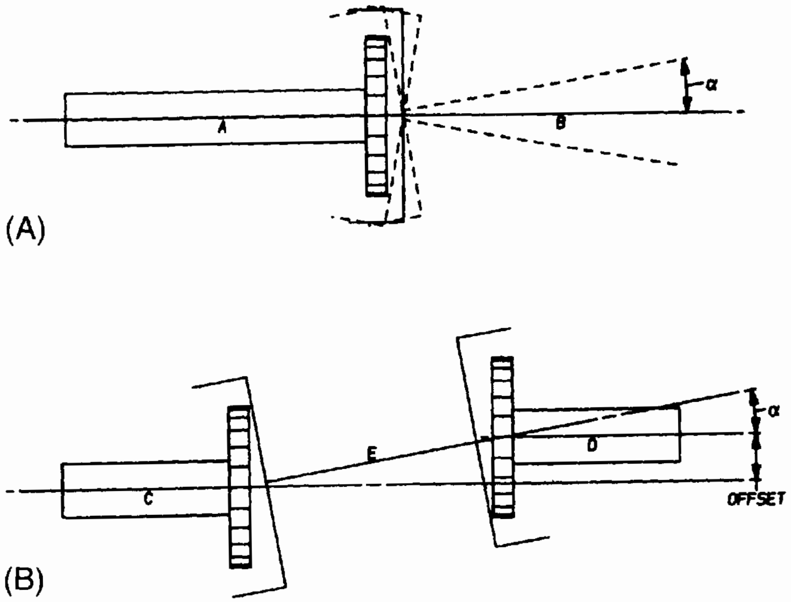


Figure 6.3 (A) Single mesh, angular misalignment. (B) Double mesh, angular and offset misalignment.

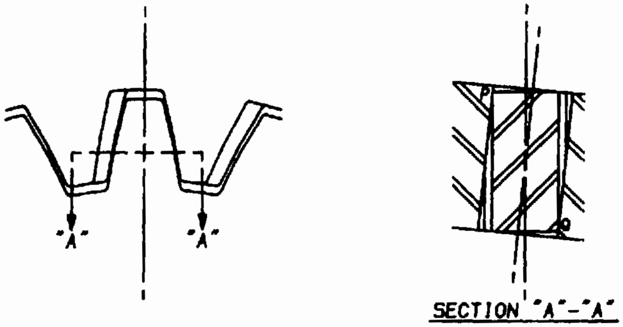


Figure 6.4 Misalignment of a conventional tooth.

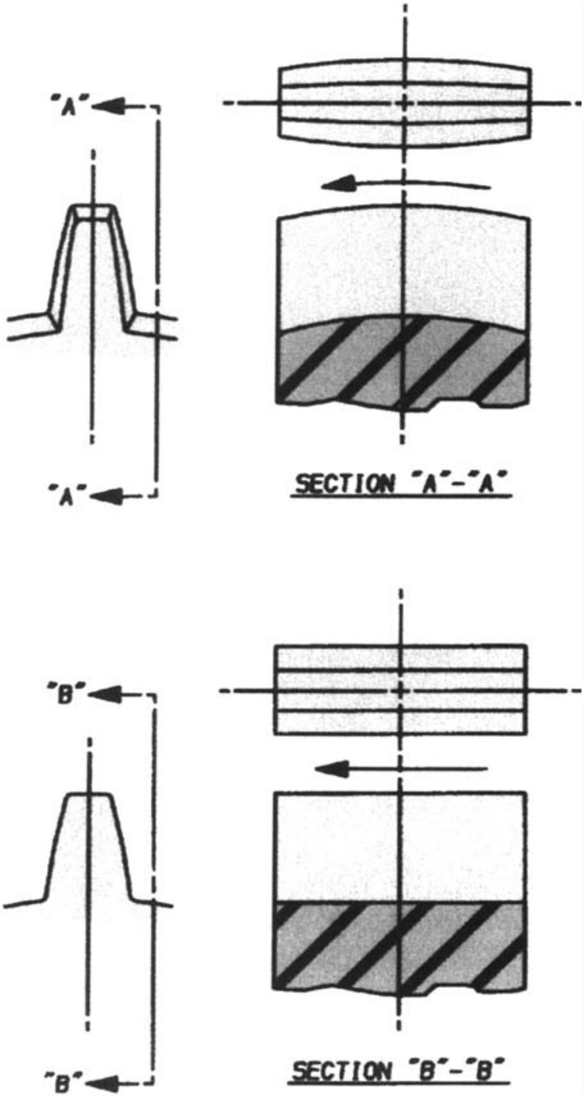


FIGURE 6.5 Path of hob for (A) crowned tooth, (B) conventional tooth.

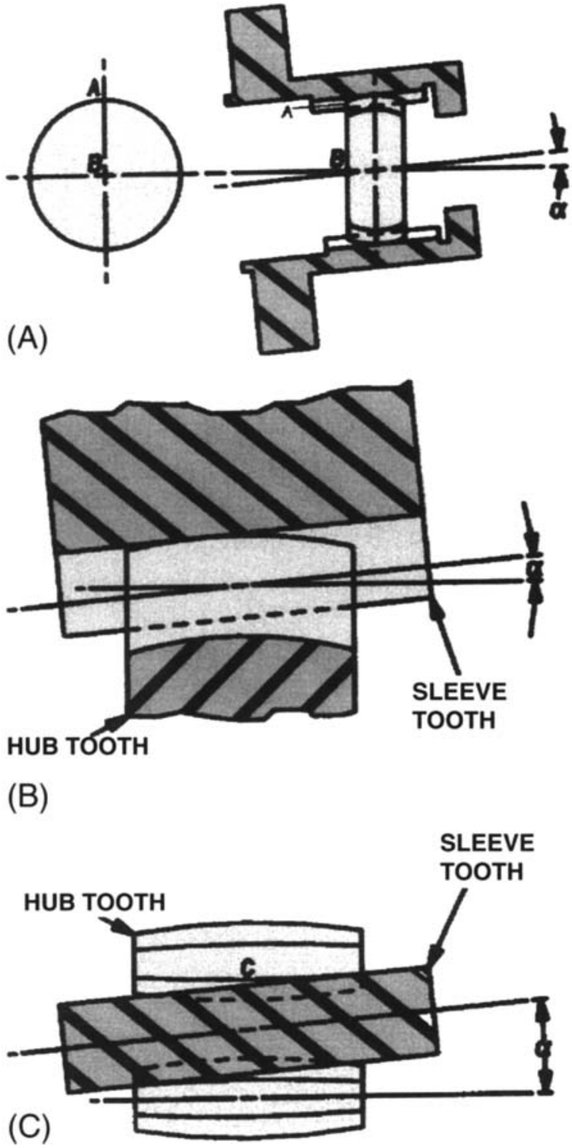


Figure 6.6 Misalignment action of a crowned tooth coupling.

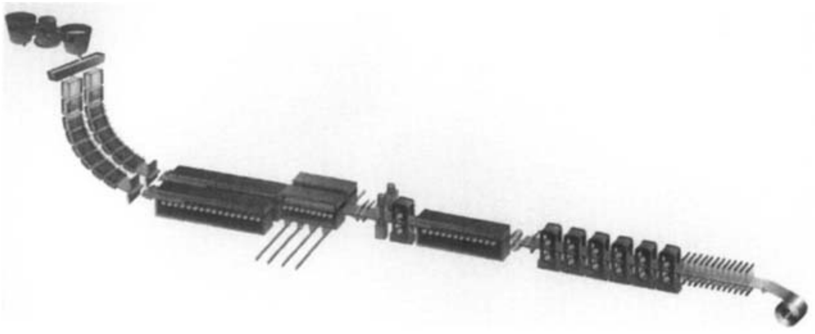


FIGURE 6.7 Compact hot strip mill. (Courtesy of Danieli-United.)

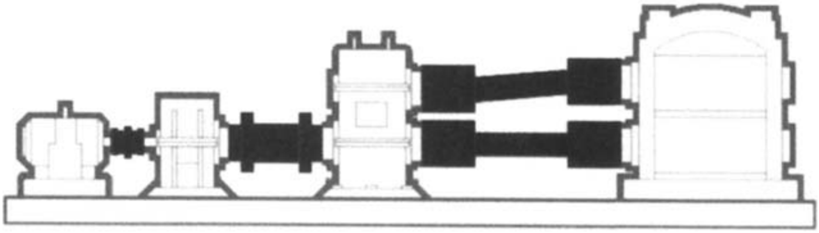


FIGURE 6.8 Typical configuration of a hot mill finishing stand.

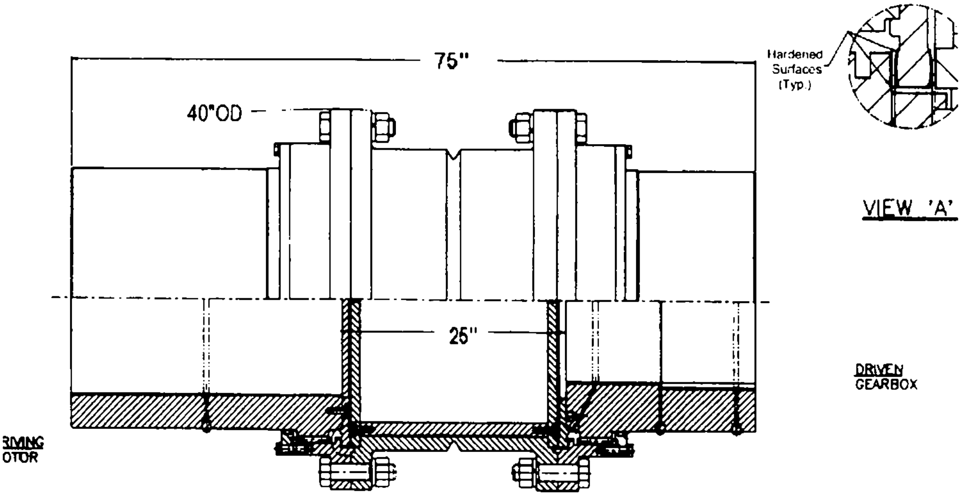


FIGURE 6.9 Typical motor-to-gear box coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

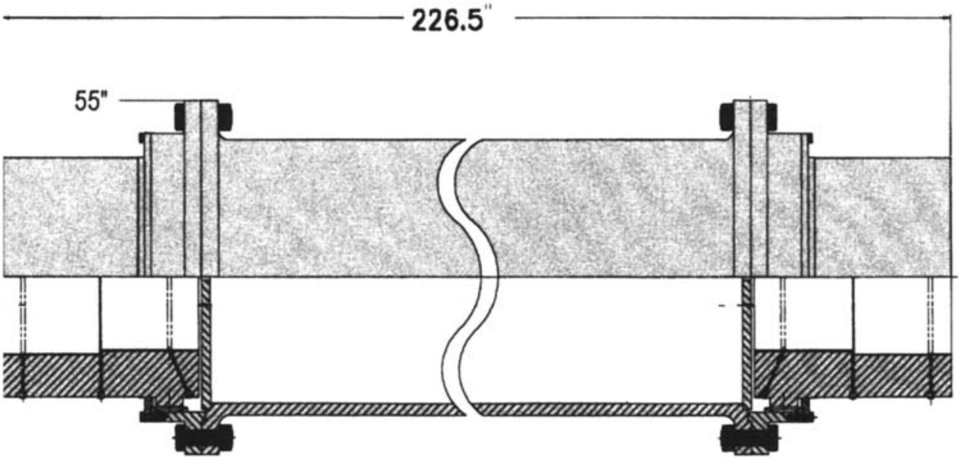


FIGURE 6.10 Gear-to-pinion coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

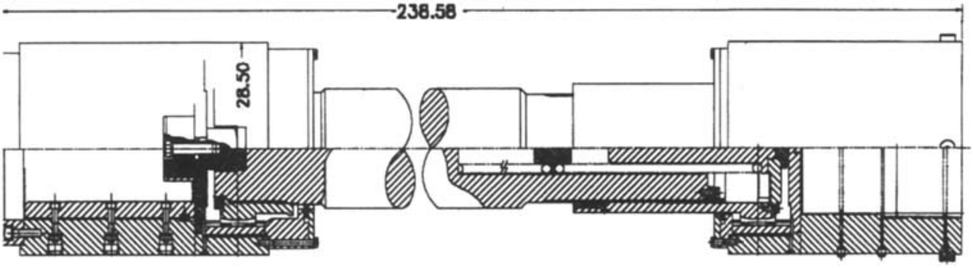


FIGURE 6.11 Gear spindle. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

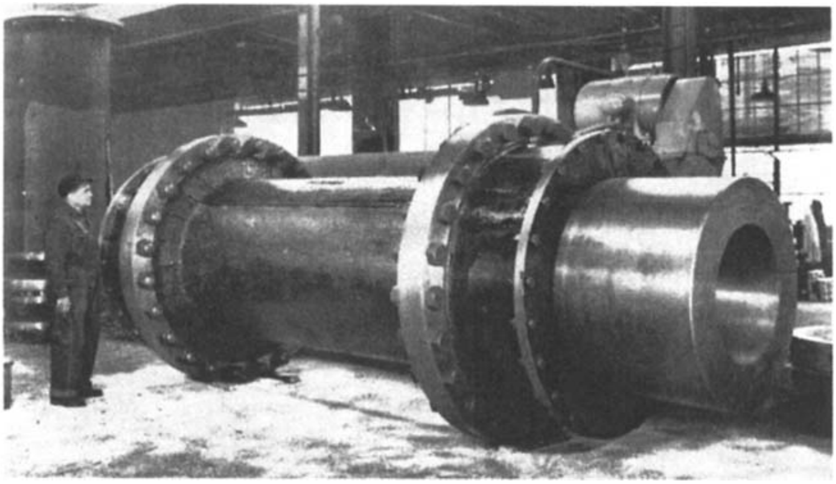


FIGURE 6.12 Large gear coupling. (Courtesy of T.B Wood's Inc. Poole.)

FIGURE 6.13 Crane drive. (Courtesy of Morgan Construction.)

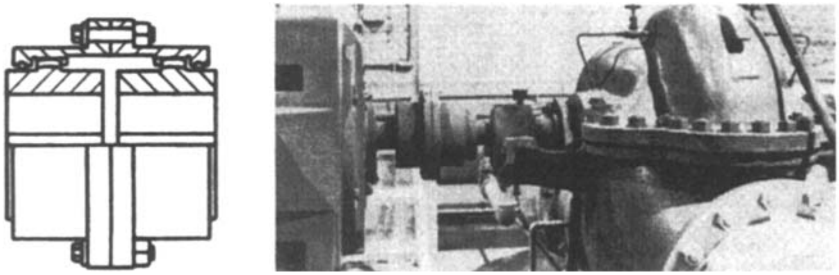


Figure 6.14 Pump drive. (Courtesy of Falk Corporation.)

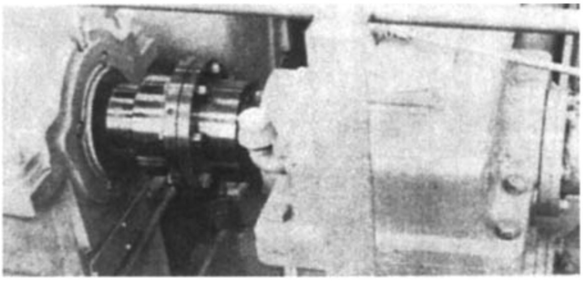


Figure 6.15 Gear drive. (Courtesy of Falk Corporation.)

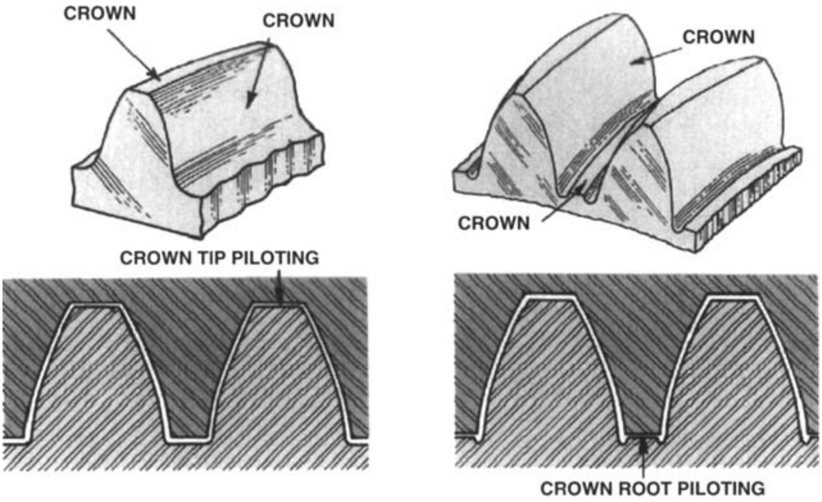


Figure 6.16 20 and 25 degree pressure angle teeth.

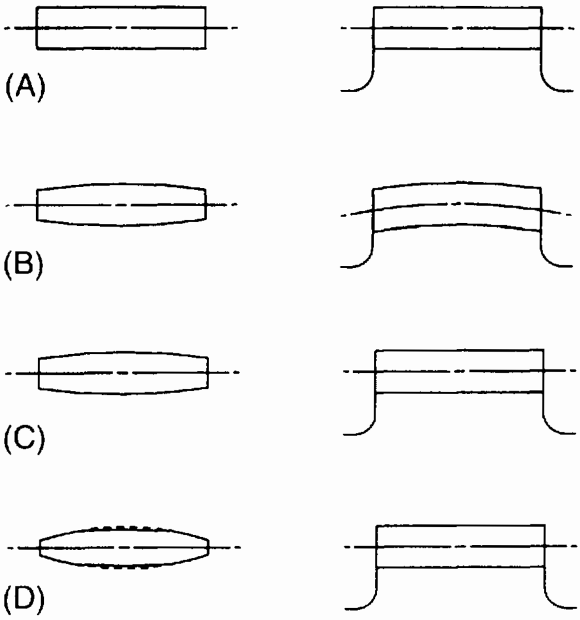


Figure 6.17 Gear coupling tooth form. (A) Straight-tooth form. (B) Crowned tooth with pitch on a curved path. (C) Crowned tooth with pitch on straight line. (D) Variable crowned tooth.

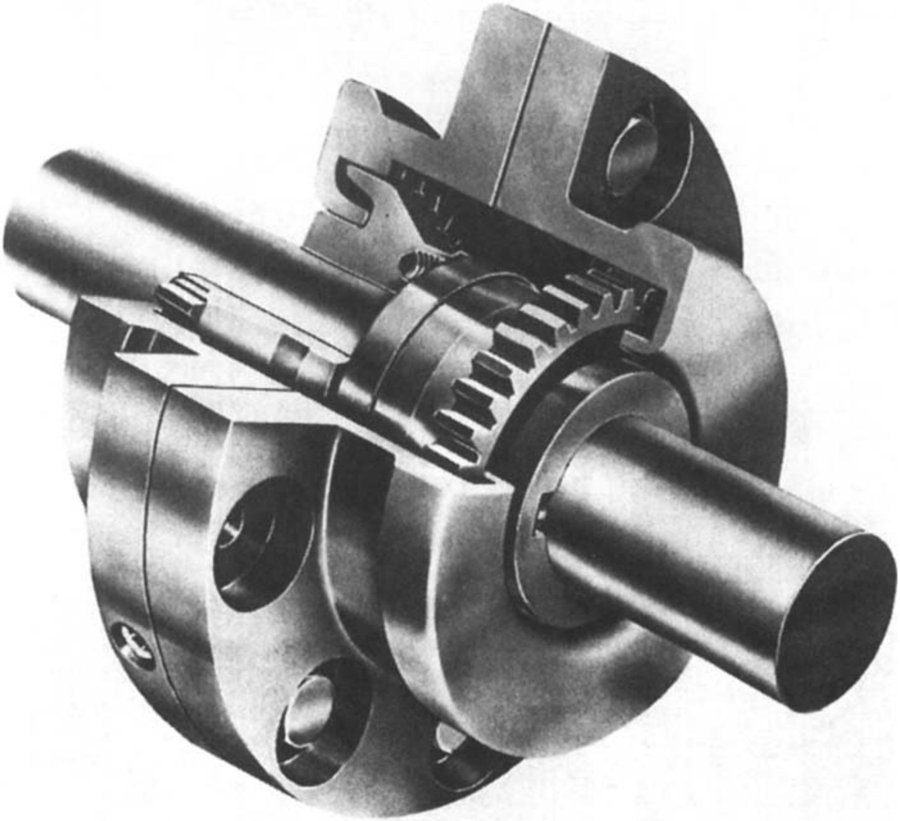


FIGURE 6.18 Straight tooth coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

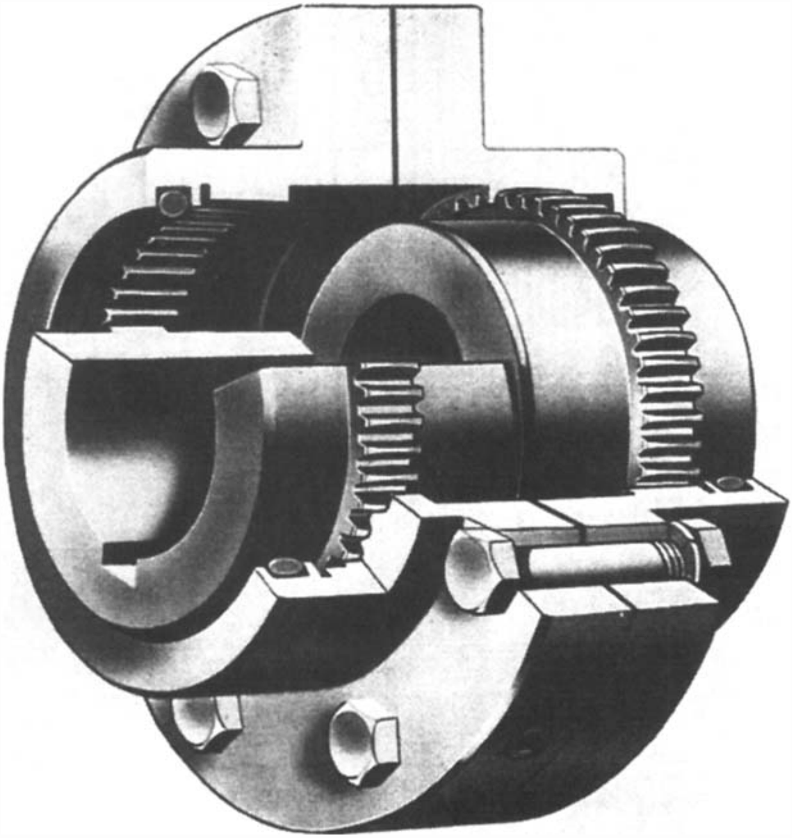
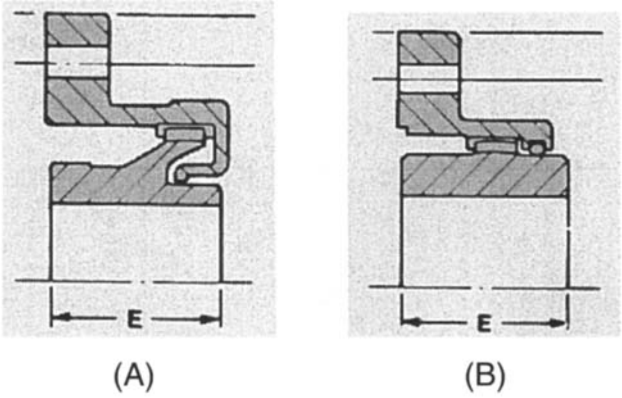


FIGURE 6.19 Crowned tooth coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)



(A) (B)

						KOP-FLEX FASTS FIGURE A			KOP-FLEX SERIES H FIGURE B			KOP-FLEX WALDRON FIGURE B			AMERIDRIVES-F-200 SERIES FIGURE B		
EXPOSED BOLTS DIMENSION S COMMON TO ALL MANUFACTURERS SHOWN HERE																	
COUPLING SIZE	OUTSIDE FLANGE DIAMETER (IN)	NUMBER OF BOLTS	BOLT SIZE (IN	BOLT CIRCLE (IN)	GAP (IN	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)
1.00	4.563	6	0.250	3.750	0.125				1.00	1.625	1.688	1.00	1.625	1.688			
1.50	6.000	8	0.375	4.812	0.125	1.50	1.625	1.938	1.50	2.250	1.938	1.50	2.188	2.062	1.50	2.250	1.938
2.00	7.000	6	0.500	5.875	0.125	2.00	2.125	2.438	2.00	2.750	2.438	2.00	2.750	2.438	2.00	2.750	2.438
2.50	8.375	6	0.625	7.125	0.188	2.50	2.750	3.030	2.50	3.500	3.030	2.50	3.250	3.030	2.50	3.500	3.030
3.00	9.438	8	0.625	8.125	0.188	3.00	3.125	3.594	3.00	4.000	3.594	3.00	4.000	3.594	3.00	4.000	3.594
3.50	11.000	8	0.750	9.500	0.250	3.50	3.750	4.188	3.50	4.500	4.188	3.50	4.750	4.188	3.50	4.500	4.188
4.00	12.500	8	0.750	11.000	0.250	4.00	4.250	4.750	4.00	5.500	4.750	4.00	5.375	4.750	4.00	5.500	4.750
4.50	13.625	10	0.750	12.000	0.313	4.50	4.750	5.313	4.50	6.000	5.313	4.50	6.000	5.375	4.50	6.250	5.313
5.00	15.313	8	0.875	13.500	0.313	5.00	5.500	6.032	5.00	6.875	6.032	5.00	6.750	6.125	5.00	7.000	6.032
5.50	16.750	14	0.875	14.500	0.313	5.50	5.875	6.906	5.50	7.750	6.906	5.50	7.500	6.625	5.50	7.500	6.625
6.00	18.000	14	0.875	15.750	0.313	6.00	6.500	7.406	6.00	8.625	7.406	6.00	8.250	7.375	6.00	8.250	7.406
7.00	20.750	16	1.000	18.250	0.375	7.00	8.000	8.688	7.00	10.375	8.688	7.00	9.250	8.688	7.00	9.620	8.688

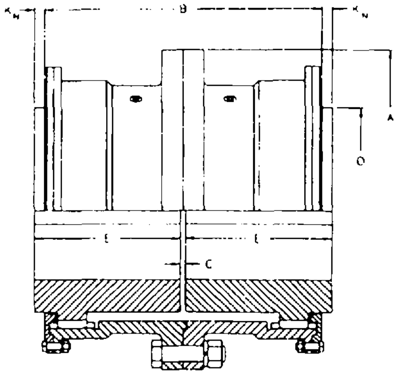
						RENOLDS-LABYRINTH METAL SEALS FIGURE A			RENOLDS-O-RING SEAL FIGURE B			LOVEJOY/SIER-BATH FLANGED SLEEVE SERIES FIGURE B			LOVEJOY/SIER-BATH ALL-METAL LABYRINTH FIGURE A		
EXPOSED BOLTS DIMENSION S COMMON TO ALL MANUFACTURERS SHOWN HERE																	
COUPLING SIZE	OUTSIDE FLANGE DIAMETER (IN)	NUMBER OF BOLTS	BOLT SIZE (IN	BOLT CIRCLE (IN)	GAP (IN	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)
1.00	4.563	6	0.250	3.750	0.125				1.00	1.625	1.688	1.00	1.625	1.688			
1.50	6.000	8	0.375	4.812	0.125	1.50	1.750	1.938	1.50	2.125	1.938	1.50	2.125	1.938	1.50	1.625	1.940
2.00	7.000	6	0.500	5.875	0.125	2.00	2.250	2.438	2.00	2.750	2.438	2.00	2.750	2.438	2.00	2.125	2.438
2.50	8.375	6	0.625	7.125	0.188	2.50	2.750	3.030	2.50	3.250	3.030	2.50	3.250	3.030	2.50	2.750	3.030
3.00	9.438	8	0.625	8.125	0.188	3.00	3.250	3.594	3.00	4.000	3.594	3.00	4.000	3.594	3.00	3.125	3.594
3.50	11.000	8	0.750	9.500	0.250	3.50	3.750	4.188	3.50	4.500	4.188	3.50	4.625	4.188	3.50	3.750	4.188
4.00	12.500	8	0.750	11.000	0.250	4.00	4.250	4.750	4.00	5.375	4.750	4.00	5.188	4.750	4.00	4.250	4.750
4.50	13.625	10	0.750	12.000	0.313	4.50	4.750	5.313	4.50	6.000	5.313	4.50	6.000	5.313	4.50	4.750	5.310
5.00	15.313	8	0.875	13.500	0.313	5.00	5.500	6.032	5.00	6.625	6.032	5.00	6.500	6.032	5.00	5.500	6.030
5.50	16.750	14	0.875	14.500	0.313	5.50	6.250	6.906	5.50	7.500	6.906	5.50	7.375	6.906	5.50	5.875	6.906
6.00	18.000	14	0.875	15.750	0.313	6.00	7.000	7.406	6.00	8.125	7.406	6.00	8.000	7.046	6.00	6.500	7.046
7.00	20.750	16	1.000	18.250	0.375	7.00	8.250	8.688	7.00	9.625	8.688	7.00	9.000	8.688	7.00	8.000	8.688

FIGURE 6.20 Comparison of standard gear couplings. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

EXPOSED BOLTS DIMENSION S COMMON TO ALL MANUFACTURERS SHOWN HERE						TB WOOD'S SERIES F FIGURE B			TB WOOD'S/POOLE MXB FIGURE B			TB WOOD'S ALL-METAL LABYRINTH SERIES K FIGURE A		
COUPLING SIZE	OUTSIDE FLANGE DIAMETER (IN)	NUMBER OF BOLTS	BOLT SIZE (IN	BOLT CIRCLE (IN)	GAP (IN	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)
1.00	4.563	6	0.250	3.750	0.125	1.00	1.625	1.688	1.00	1.625	1.688			
1.50	6.000	8	0.375	4.812	0.125	1.50	2.125	1.938	1.50	2.188	2.062	1.50	1.625	1.940
2.00	7.000	6	0.500	5.875	0.125	2.00	2.750	2.438	2.00	2.750	2.437	2.00	2.125	2.438
2.50	8.375	6	0.625	7.125	0.188	2.50	3.250	3.030	2.50	3.250	3.030	2.50	2.750	3.030
3.00	9.438	8	0.625	8.125	0.188	3.00	4.000	3.594	3.00	4.000	3.594	3.00	3.125	3.594
3.50	11.000	8	0.750	9.500	0.250	3.50	4.625	4.188	3.50	4.750	4.188	3.50	3.750	4.188
4.00	12.500	8	0.750	11.000	0.250	4.00	5.188	4.750	4.00	5.375	4.750	4.00	4.250	4.750
4.50	13.625	10	0.750	12.000	0.313	4.50	6.000	5.313	4.50	6.000	5.375	4.50	4.750	5.310
5.00	15.313	8	0.875	13.500	0.313	5.00	6.500	6.032	5.00	6.750	6.125	5.00	5.500	6.030
5.50	16.750	14	0.875	14.500	0.313	5.50	7.375	6.906	5.50	7.500	6.625	5.50	5.875	6.906
6.00	18.000	14	0.875	15.750	0.313	6.00	8.000	7.046	6.00	8.250	7.375	6.00	6.500	7.046
7.00	20.750	16	1.000	18.250	0.375	7.00	9.000	8.688	7.00	9.250	8.688	7.00	8.000	8.688

EXPOSED BOLTS DIMENSION S COMMON TO ALL MANUFACTURERS SHOWN HERE						SYSTEM COMPONENTS POWERTORK SERIES F FIGURE B			SYSTEM COMPONENTS STEEL RING COUPLING FIGURE A			FALK G SERIES FIGURE B		
COUPLING SIZE	OUTSIDE FLANGE DIAMETER (IN)	NUMBER OF BOLTS	BOLT SIZE (IN	BOLT CIRCLE (IN)	GAP (IN	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)	SIZE	MAX BORE (IN)	BORE LENGTH E (IN)
1.00	4.563	6	0.250	3.750	0.125	1.00	1.620	1.690				1.00	1.875	1.688
1.50	6.000	8	0.375	4.812	0.125	1.50	2.190	1.938	1.50	1.625	1.938	1.50	2.375	1.938
2.00	7.000	6	0.500	5.875	0.125	2.00	2.750	2.438	2.00	2.125	2.438	2.00	2.875	2.438
2.50	8.375	6	0.625	7.125	0.188	2.50	3.500	3.030	2.50	2.750	3.030	2.50	3.625	3.030
3.00	9.438	8	0.625	8.125	0.188	3.00	4.000	3.594	3.00	3.125	3.594	3.00	4.125	3.594
3.50	11.000	8	0.750	9.500	0.250	3.50	4.625	4.188	3.50	3.750	4.188	3.50	4.875	4.188
4.00	12.500	8	0.750	11.000	0.250	4.00	5.500	4.750	4.00	4.250	4.750	4.00	5.750	4.750
4.50	13.625	10	0.750	12.000	0.313	4.50	6.000	5.313	4.50	4.750	5.313	4.50	6.750	5.313
5.00	15.313	8	0.875	13.500	0.313	5.00	6.880	6.032	5.00	5.500	6.032	5.00	7.375	6.032
5.50	16.750	14	0.875	14.500	0.313	5.50	7.750	6.625	5.50	5.875	6.906	5.50	8.250	6.625
6.00	18.000	14	0.875	15.750	0.313	6.00	8.750	7.046	6.00	6.500	7.046	6.00	9.125	7.406
7.00	20.750	16	1.000	18.250	0.375	7.00	10.375	8.688	7.00	8.000	8.688	7.00	10.875	8.688

FIGURE 6.20 CONTINUED



Coupling Size	Maximum Bore with Standard Key	Rating HP / 100 RPM	Torque Rating (lb.-in. x 1000)	Peak Torque Rating (lb.-in. x 1000)	Maximum Speed (RPM)	Weight with Solid Hubs (lb.)	Dimensions						
							A	B	C	C_1	C_W	E	O
8	10 3/4	2230	1404	2808	1750	1430	23 1/4	19 1/2	3/8	5 9/16	10 3/4	9 13/16	14
9	11 3/4	3170	1995	3990	1625	2000	26	21 3/4	1/2	6 1/4	12	10 7/8	15 1/2
10	13	4360	2744	5488	1500	2670	28	24	1/2	7 1/8	13 3/4	12	17 1/2
11	15	5780	3645	7290	1375	3520	30 1/2	26 1/4	1/2	7 7/8	15 1/4	13 1/8	19 1/2
12	16 1/4	7190	4532	9064	1250	4450	33	27 3/4	1/2	8 1/8	15 3/4	13 7/8	21 1/2
13	17 1/2	9030	5688	11376	1125	5410	35 3/4	29 1/2	3/4	8 7/16	16 1/8	14 5/8	23
14	18 3/4	11080	6982	13964	1000	6800	38	31 1/4	3/4	9	17 1/4	15 1/2	25
15	20 3/4	13470	8488	16976	875	8040	40 1/2	33 1/8	3/4	9 7/16	18 1/8	16 1/2	27
16	22	16100	10150	20300	750	9680	43	35	1	9 7/8	18 3/4	17 3/8	29
18	25 1/4	21100	13300	26600	500	12500	47 1/4	36 1/8	1	10 1/4	19 1/2	18	33
20	27 1/4	28800	18144	36286	400	17900	53 1/2	42 3/8	1	12 3/8	23 3/4	21 1/8	36 1/2
22	30	38100	24009	48018	300	23300	59	48	1	13 3/4	26 1/2	23	40
24	33 1/4	42400	26699	53398	200	30300	64 1/4	49 1/2	1	15	29	24 3/4	44 1/2
26	36 3/4	53000	33415	66830	200	37700	68 1/2	53	1	15 5/8	30 1/4	26 1/2	48 1/2
28	40	65900	41564	83128	200	45200	73 3/4	54 1/4	1	15 7/8	30 3/4	27 1/8	52 1/2
30	43 1/2	80300	50614	101228	200	52700	78	55 1/4	1	15 7/8	30 3/4	27 5/8	56 1/2

FIGURE 6.21 Typical dimensions for large size gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

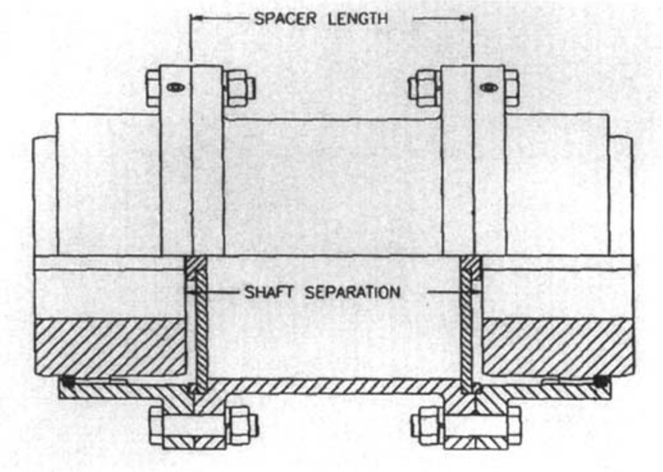


FIGURE 6.22 Spacer type gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

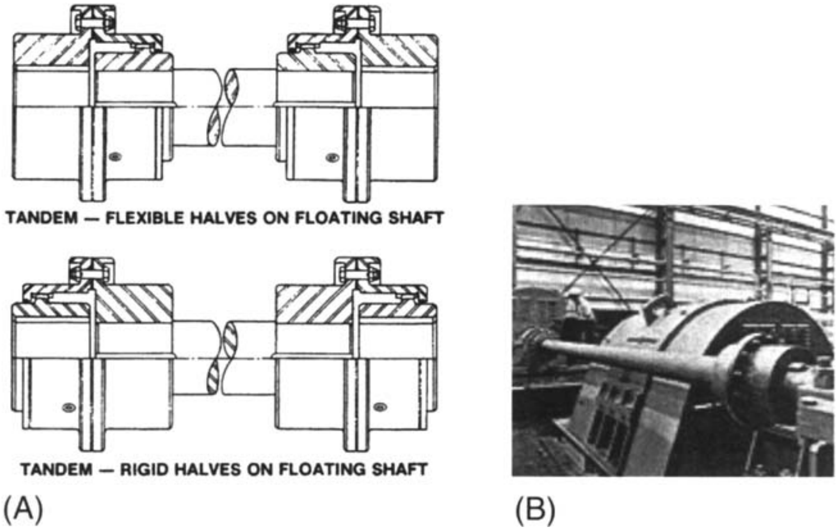

TANDEM — FLEXIBLE HALVES ON FLOATING SHAFT
TANDEM — RIGID HALVES ON FLOATING SHAFT
(A)
(B)

FIGURE 6.23 (A) Solid-shaft gear coupling. (B) Solid-shaft steel mill application. (Courtesy of Ameridrives International.)

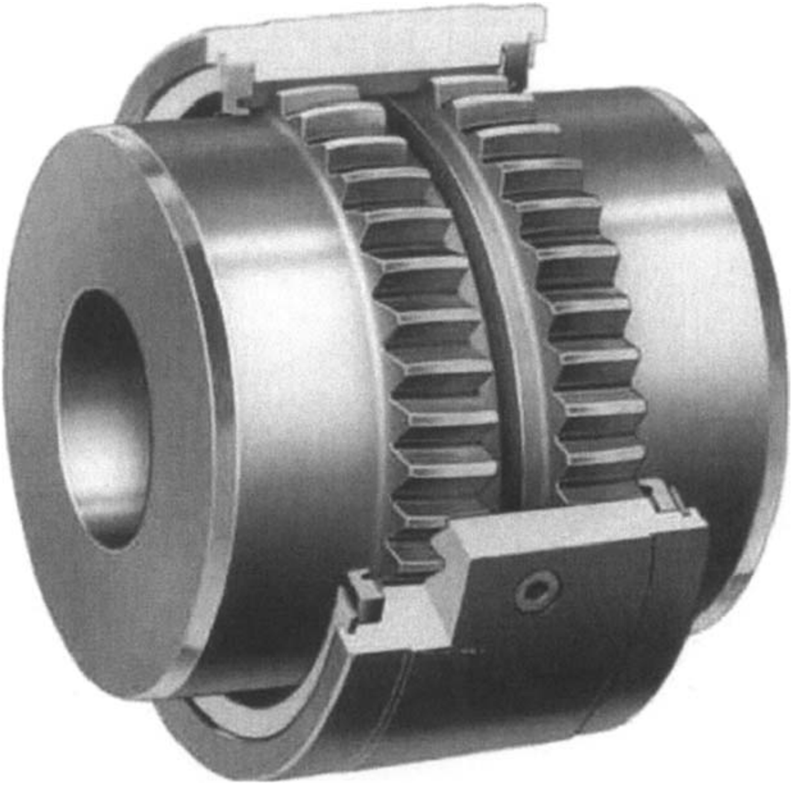


FIGURE 6.24 Flangeless gear coupling. (Courtesy of Kop-Flex, Inc.,

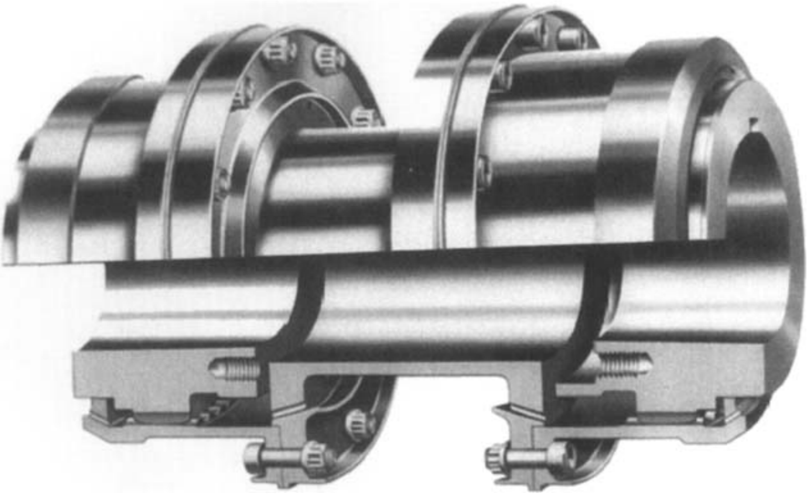
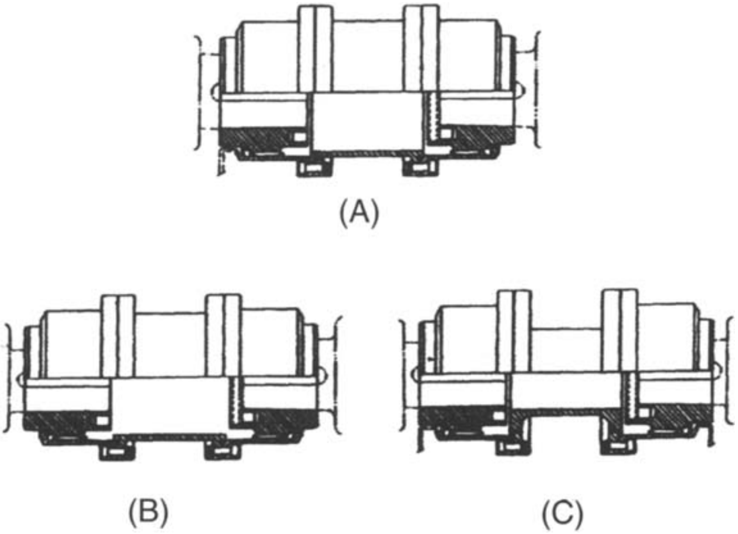


FIGURE 6.25 Typical high-speed gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)



(A)

(B)

(C)

FIGURE 6.26 Types of high-speed gear couplings. (A) Spacer coupling for flow-through continuous lube. (B) Spacer coupling for packed lube. (C) Spacer coupling for dual direction continuous lube (end-lubed).

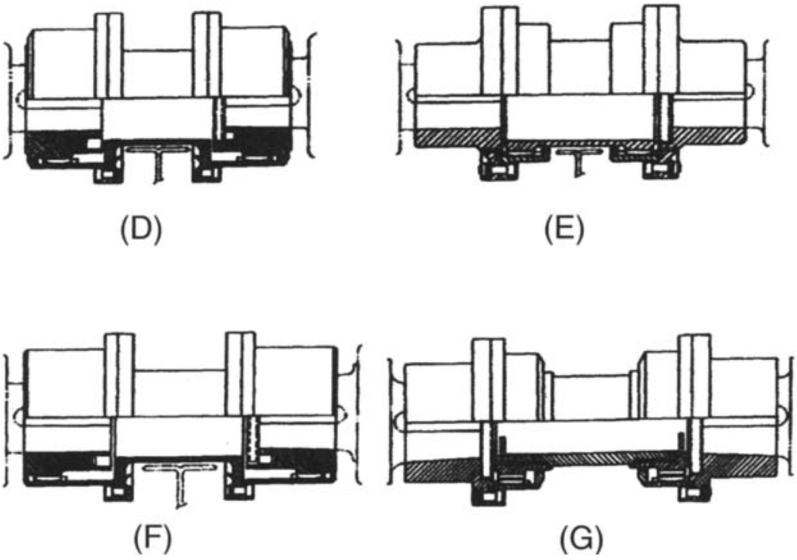


Figure 6.26 Continued. (D) Spacer coupling for dual direction continuous lube (center-lubed). (E) Spool-type coupling for dual direction continuous lube. (F) Spacer coupling for dual-direction continuous lube with reduced over hung moment. (G) Spool-type coupling for packed lube. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

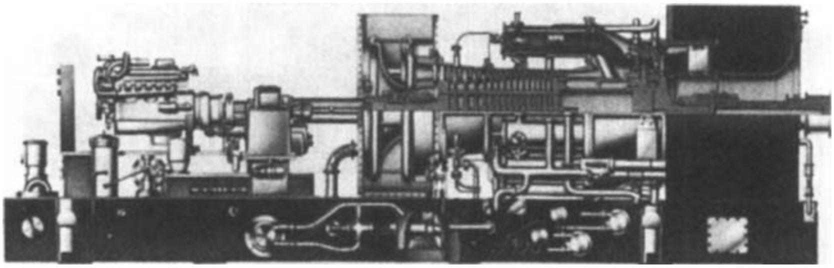


Figure 6.27 Gas turbine application. (Courtesy of General Electric Schenectady, Gas Turbine Division.)

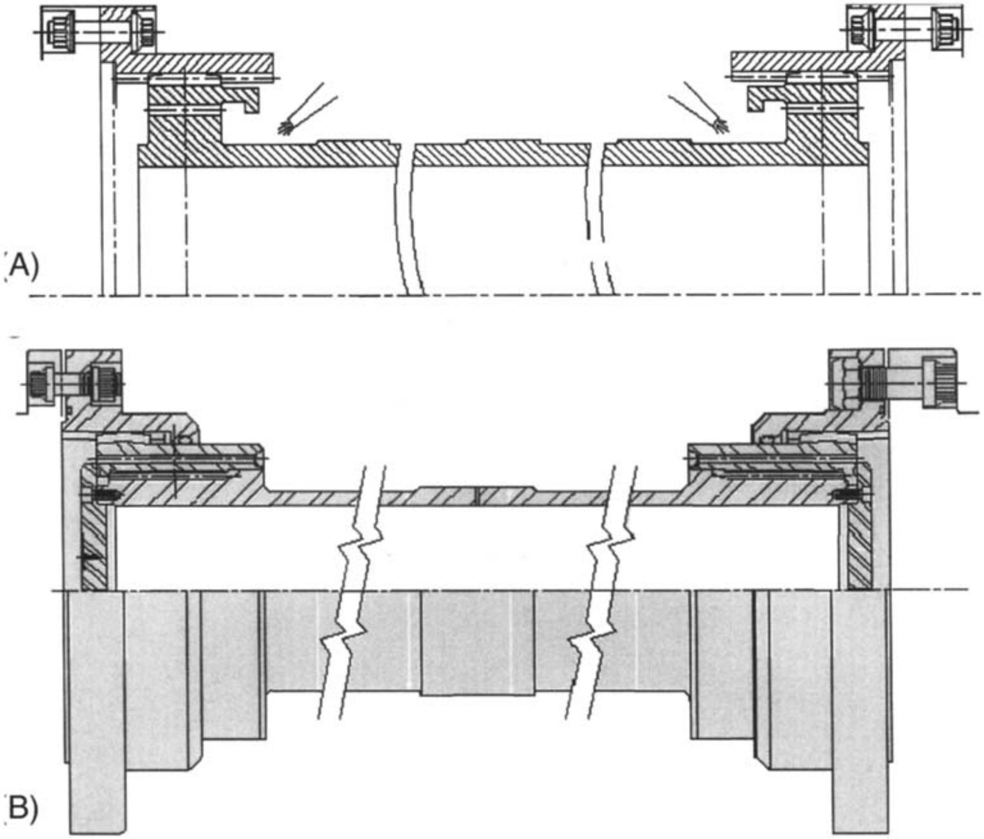


FIGURE 6.28 Typical gas turbine couplings. (A) Turbine load coupling (mechanical drive or generator). (B) Accessory drive coupling.

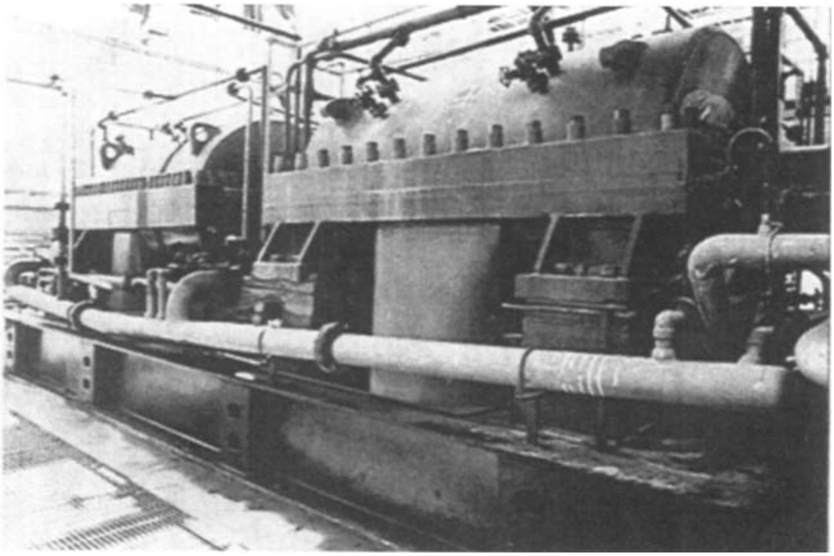


FIGURE 6.29 Centrifugal compressor application. (Courtesy of Dresser-Rand.)

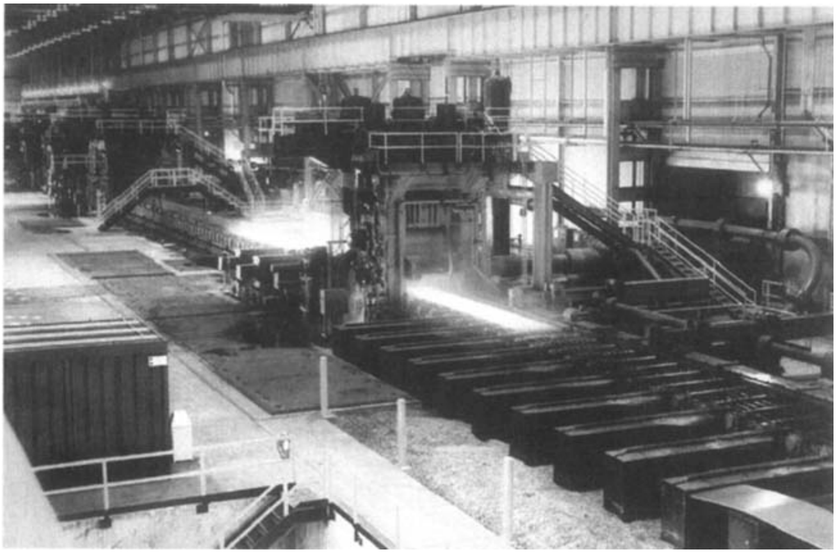
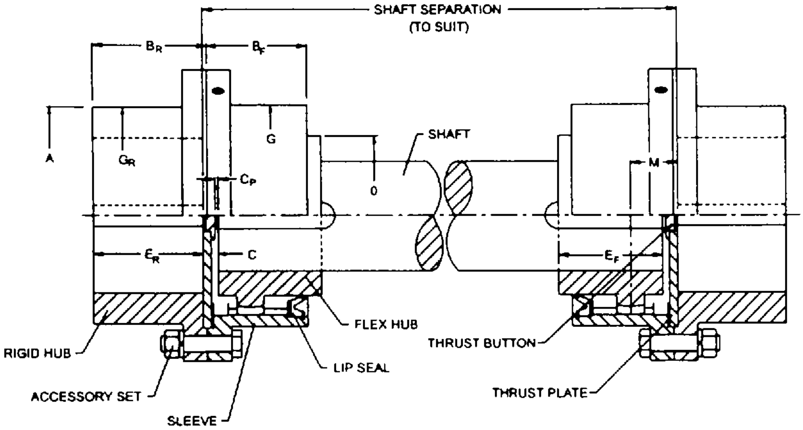


FIGURE 6.30 Hot strip mill.



Model 6° SF Specifications (inches)

Size	Maximum Bore with Standard Keyway		Dimensions										
	Flex	Rigid	A	B_F	B_R	C	C_P	D	E_F	E_R	G	G_R	O
1.5	1.875	2.625	6.00	2.22	1.94	0.34	0.06	2.00	2.31	1.84	3.94	3.81	2.62
2.0	2.375	3.250	7.00	2.62	2.38	0.37	0.09	2.50	2.81	2.28	4.94	4.81	3.38
2.5	2.875	4.000	8.38	2.78	3.00	0.50	0.12	3.00	3.03	2.91	5.88	5.75	4.00
3.0	3.500	4.750	9.44	3.31	3.56	0.50	0.12	3.75	3.59	3.47	6.88	6.75	4.88
3.5	4.000	5.500	11.00	3.97	4.12	0.53	0.15	4.25	3.94	4.03	8.00	7.75	5.62
4.0	4.625	6.250	12.50	4.31	4.62	0.72	0.15	4.75	4.38	4.44	9.25	9.00	6.50
4.5	5.375	7.250	13.63	4.86	5.25	0.72	0.16	5.50	4.75	5.06	10.38	10.12	7.50
5.0	6.000	8.500	15.31	5.28	5.88	0.75	0.19	6.00	5.50	5.69	11.56	11.38	8.50
5.5	6.625	8.000	16.75	6.31	7.16	0.82	0.19	7.00	6.50	6.97	12.69	10.75	9.25
6.0	7.125	8.750	18.00	7.00	7.66	0.88	0.25	7.50	7.25	7.47	13.88	11.50	10.00
7.0	8.500	10.000	20.75	8.00	9.00	1.06	0.25	8.00	8.38	8.75	16.06	13.38	12.00

Figure 6.31 Typical design and dimensions for flange type gear spindle. (Courtesy of Ameridrives International.)

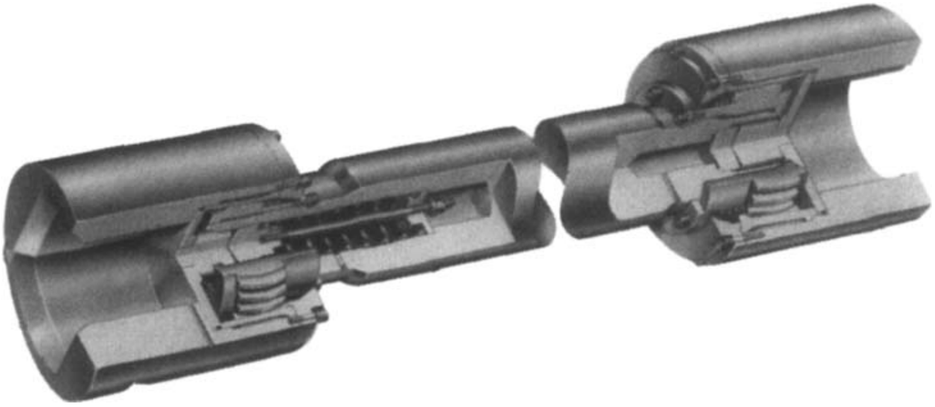
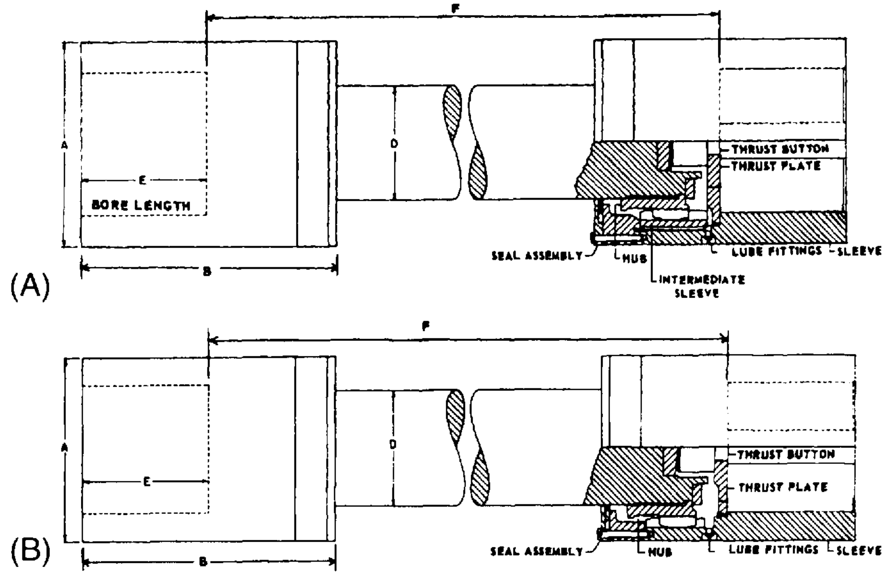


Figure 6.32 Typical gear spindle for primary mill roll drives. [Courtesy of Renolds (Ajax).]



CPLG SIZE			Typical Dimensions Sleeves and Bore Sizes						Typical Common to Both Designs LE/LB			
By OD WO/Gear Ring	By OD W/Gear Ring	By Gear Pitch Radius (in)	LB A (in)	Max Bore Roll End (in)	Max Bore Pinion end (in)	LE A (in)	Max Bore Roll End (in)	Max Bore Pinion end (in)	LE/LB B (in)	LE/LB D (in)	LE/LB E (in)	LE/LB F(Typical) (in)
10.00	11.00	4.0	10.00	8.00	7.14	11.00	8.80	7.86	13.50	6.00	6.75	68.00
11.25	12.50	4.5	11.25	9.00	8.04	12.50	10.00	8.93	15.00	6.75	7.50	75.00
12.50	14.00	5.0	12.50	10.00	8.93	14.00	11.20	10.00	17.50	7.50	8.75	90.00
13.88	15.38	5.7	13.88	11.10	9.91	15.38	12.30	10.99	19.25	8.25	9.63	100.00
14.75	16.25	6.0	14.75	11.80	10.54	16.25	13.00	11.61	20.50	8.75	10.25	100.00
15.75	17.25	6.4	15.75	12.60	11.25	17.25	13.80	12.32	21.50	9.00	10.75	110.00
16.50	18.25	6.8	16.50	13.20	11.79	18.25	14.60	13.04	23.25	9.75	11.63	120.00
17.75	19.63	7.2	17.75	14.20	12.68	19.63	15.70	14.02	24.50	10.50	12.25	120.00
19.00	20.88	7.8	19.00	15.20	13.57	20.88	16.70	14.91	26.25	11.25	13.13	130.00
19.63	21.50	8.0	19.63	15.70	14.02	21.50	17.20	15.36	27.50	11.75	13.75	140.00
20.88	23.25	8.5	20.88	16.70	14.91	23.25	18.60	16.61	29.00	12.50	14.50	150.00
22.00	24.75	9.0	22.00	17.60	15.71	24.75	19.80	17.68	30.50	13.00	15.25	150.00
23.50	25.75	9.5	23.50	18.80	16.79	25.75	20.60	18.39	32.00	14.00	16.00	150.00
24.63	27.00	10.0	24.63	19.70	17.59	27.00	21.60	19.29	34.00	15.00	17.00	150.00
26.25	30.00	10.8	26.25	21.00	18.75	30.00	24.00	21.43	36.50	17.00	18.25	150.00
27.12	31.00	11.5	27.12	21.70	19.37	31.00	24.80	22.14	39.00	18.00	19.50	150.00
28.50	32.00	12.0	28.50	22.80	20.36	32.00	25.60	22.86	40.25	19.00	20.13	150.00
29.75	34.00	12.7	29.75	23.80	21.25	34.00	27.20	24.29	42.50	20.00	21.25	150.00
32.50	37.00	14.0	32.50	26.00	23.21	37.00	29.60	26.43	47.00	21.00	23.50	150.00
34.00	38.00	15.0	34.00	27.20	24.29	38.00	30.40	27.14	47.75	22.50	23.88	150.00

Figure 6.33 Typical dimensions for large gear spindles. (A) LB spindle with gear ring. (B) LB spindle without gear ring. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

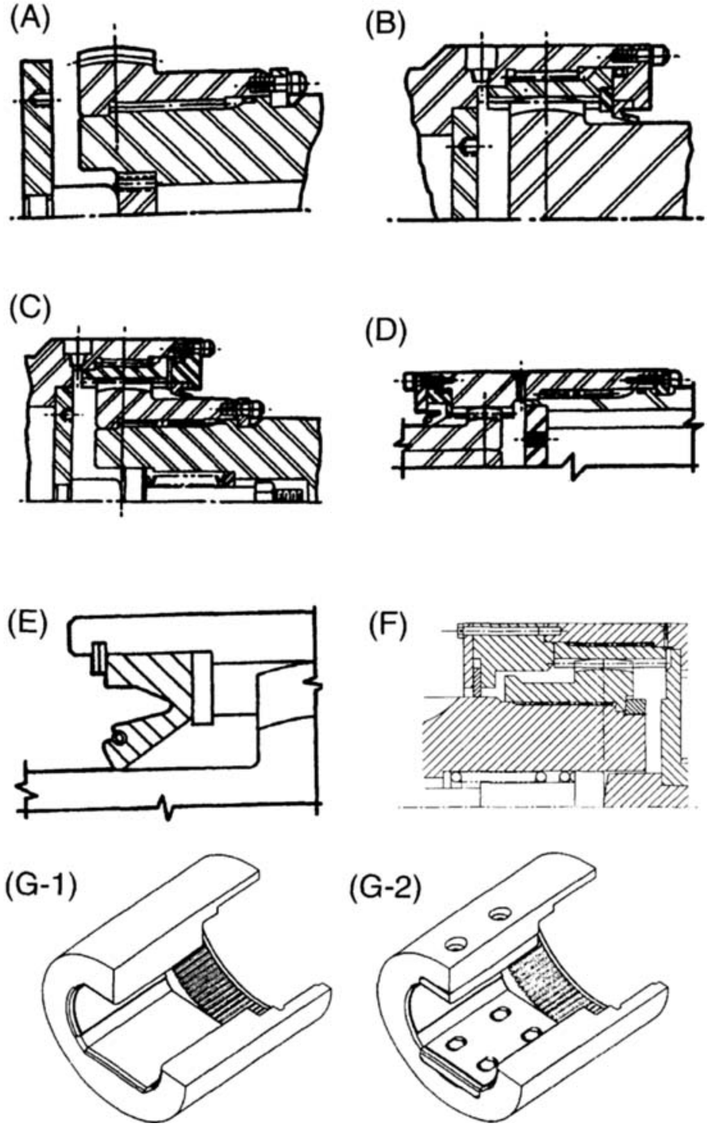
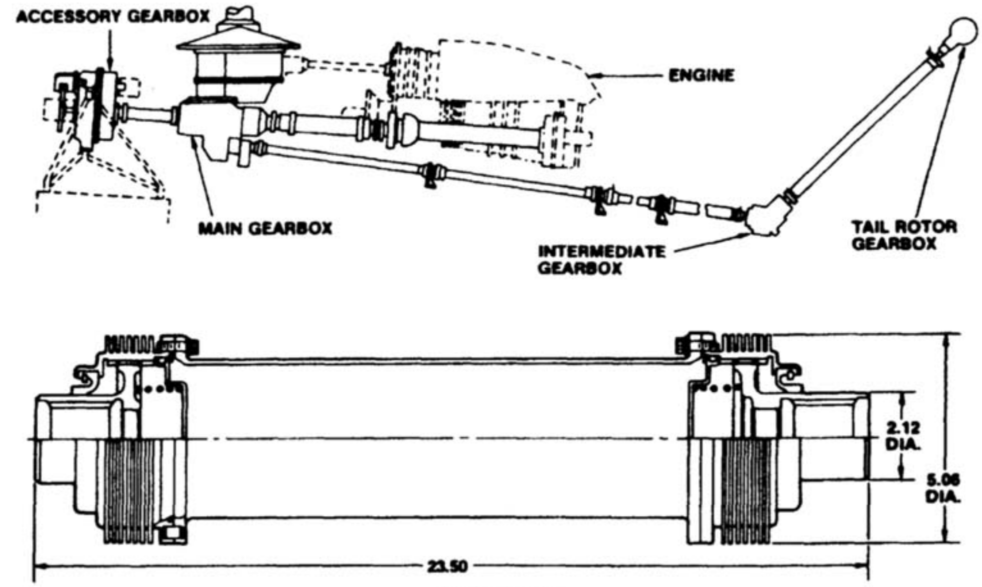
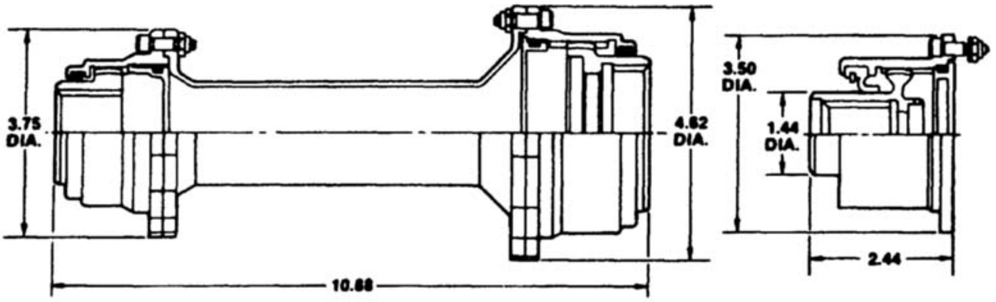


Figure 6.34 Various features of gear spindles. (A) Splined hubs. (B) Intermediate gear rings. (C) Spring-loaded thrust button. (D) Splined replaceable sleeve (E) Lip seals (F) Rising ring seal (G) (*Left*) Roll and bore-shaped bore. (*Right*) Roll end bore-with key inserts.



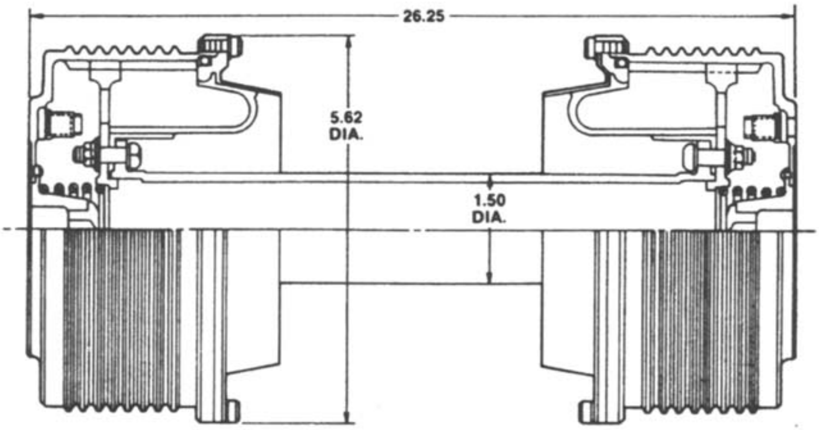
1 MAIN DRIVE



2 ACCESSORY DRIVE

3 TAIL ROTOR DRIVE

Figure 6.35 Helicopter drive train and couplings. (Courtesy of Ameridrives International.)



HELICOPTER MAIN DRIVE BETWEEN ENGINE AND TRANSMISSION. 305 HP @ 6,000 RPM. CONTINUOUS MISALIGNMENT 2.5°. INTERMITTENT MISALIGNMENT 4.5°. COUPLING MADE OF NITRALLOY N. NITRIDED LAPPED TEETH.

Figure 6.36 (*Top*) Helicopter main drive coupling. (Courtesy of Ameridrives International). (*Bottom*) Helicopter (Courtesy of Soloy Aviation).

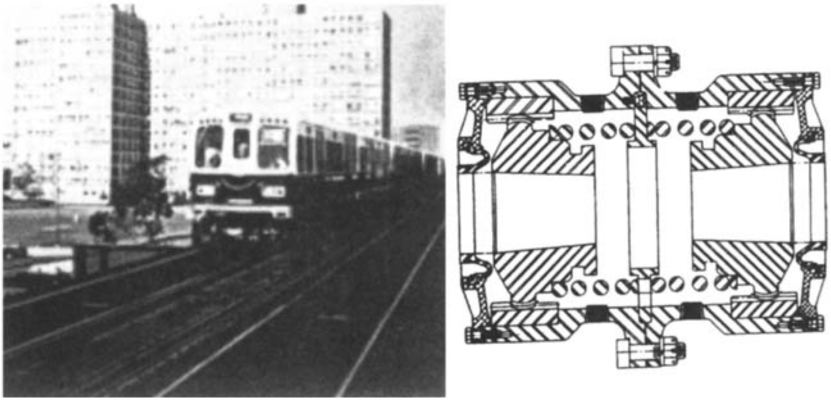


FIGURE 6.37 Traction drive coupling. (Courtesy of Ameridrives International.)

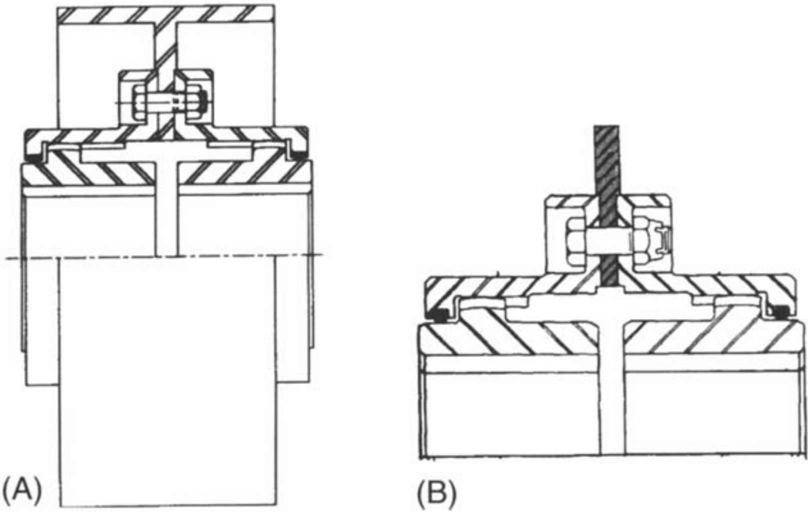


FIGURE 6.38 (A) Brake drum gear coupling. (B) Brake disk gear coupling.

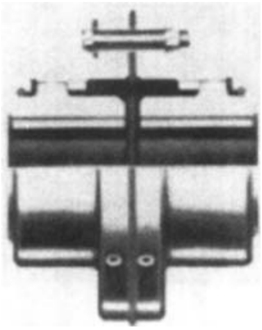


FIGURE 6.39 Insulated gear coupling. (Courtesy of Ameridrives International.)

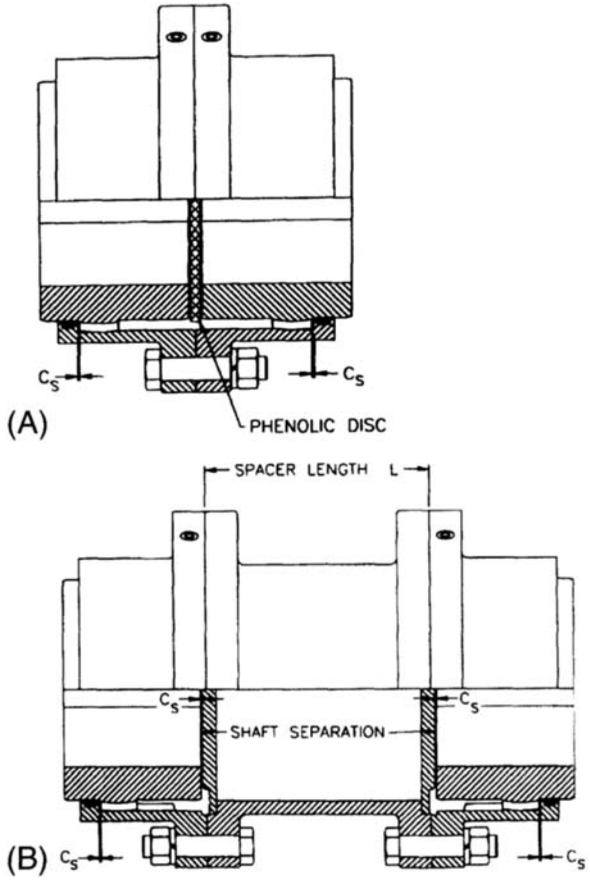


FIGURE 6.40 (A) Limited end float close-coupled gear coupling. (B) Limited end float spacer gear coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

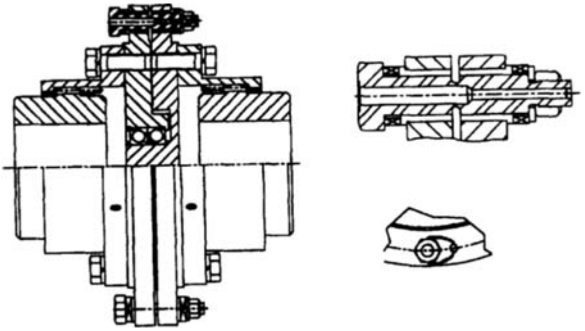


FIGURE 6.41 Shear cartridge gear coupling. (Courtesy of Kop-Flex, Emerson Power Transmission Corporation.)

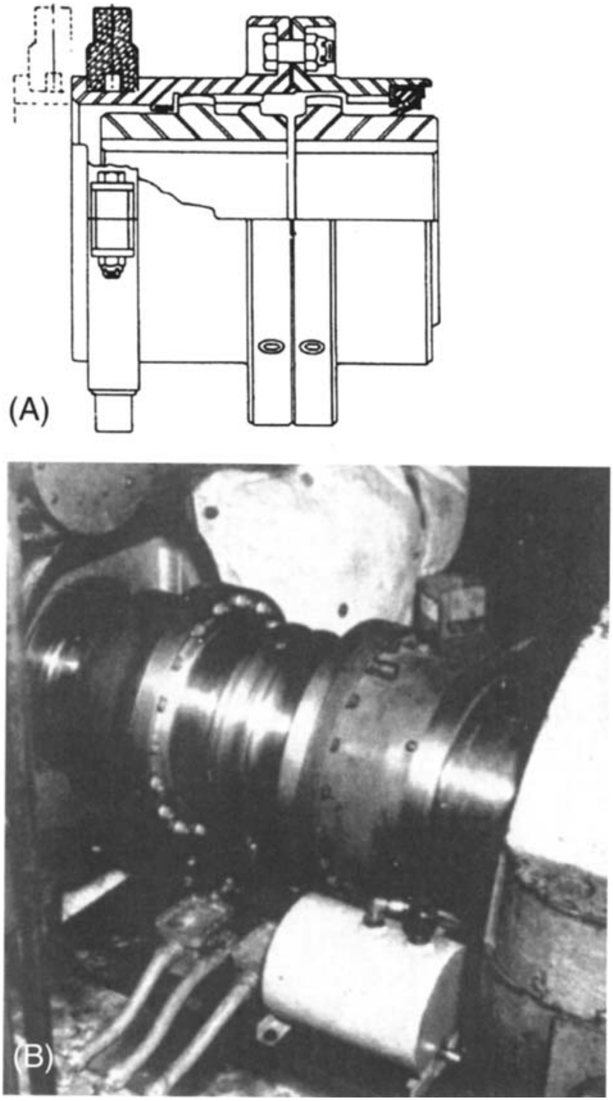


Figure 6.42 Disconnect gear couplings. (A) Low speed. (B) High speed. (Courtesy of Ameridrives International.)

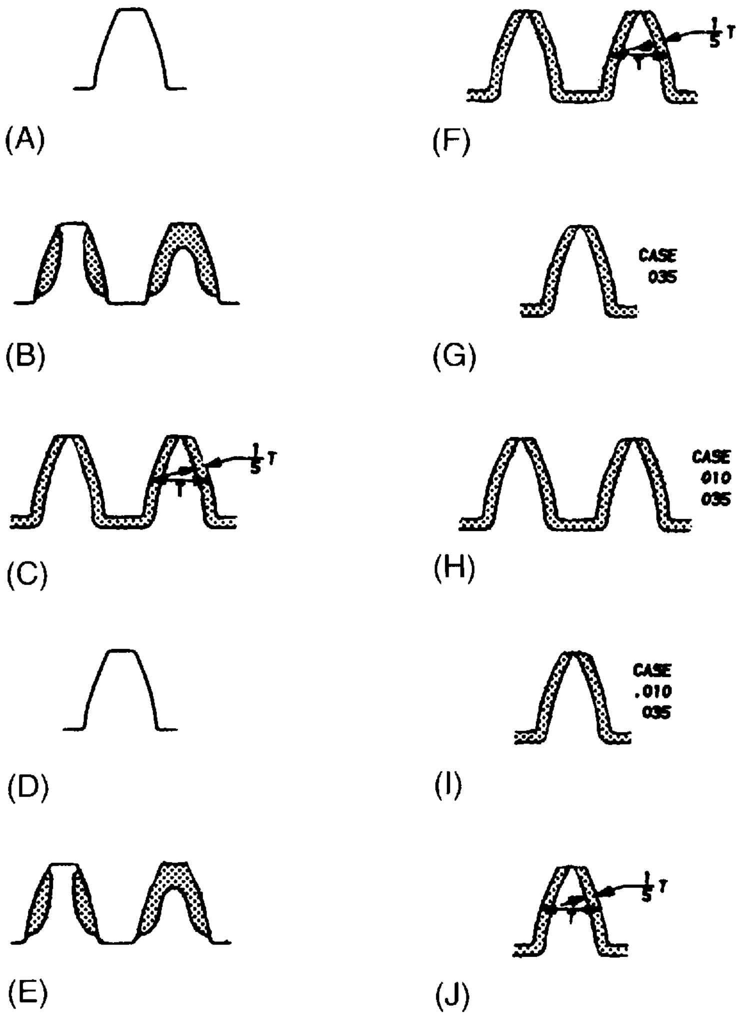
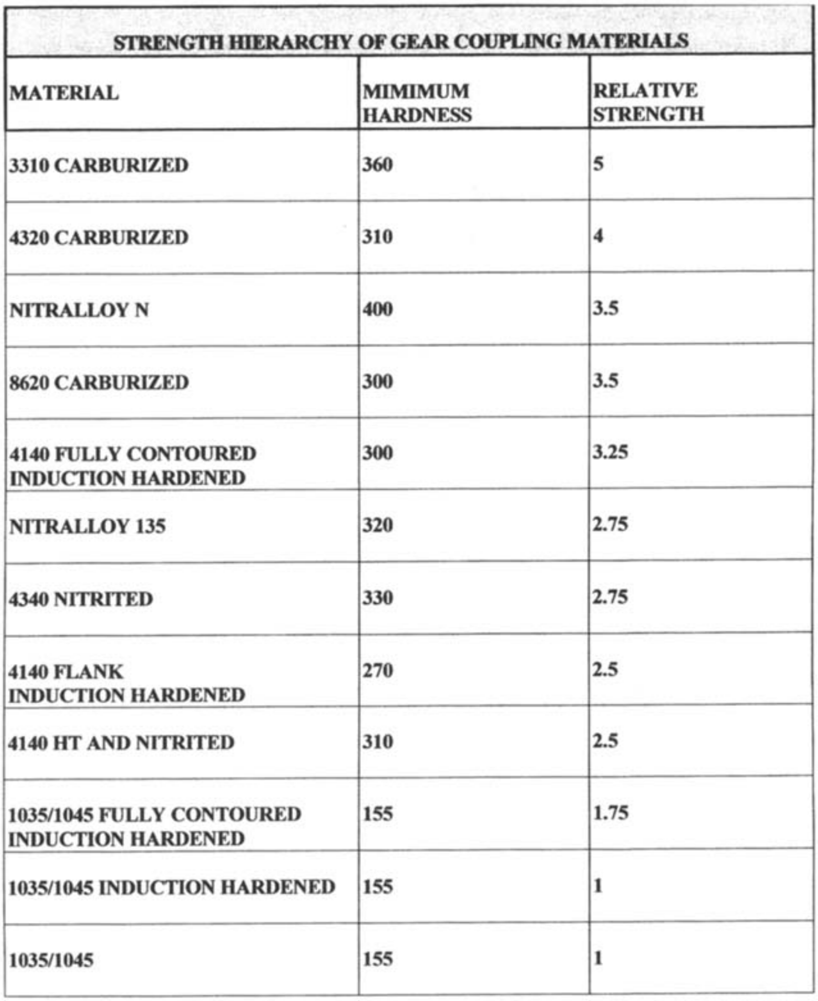


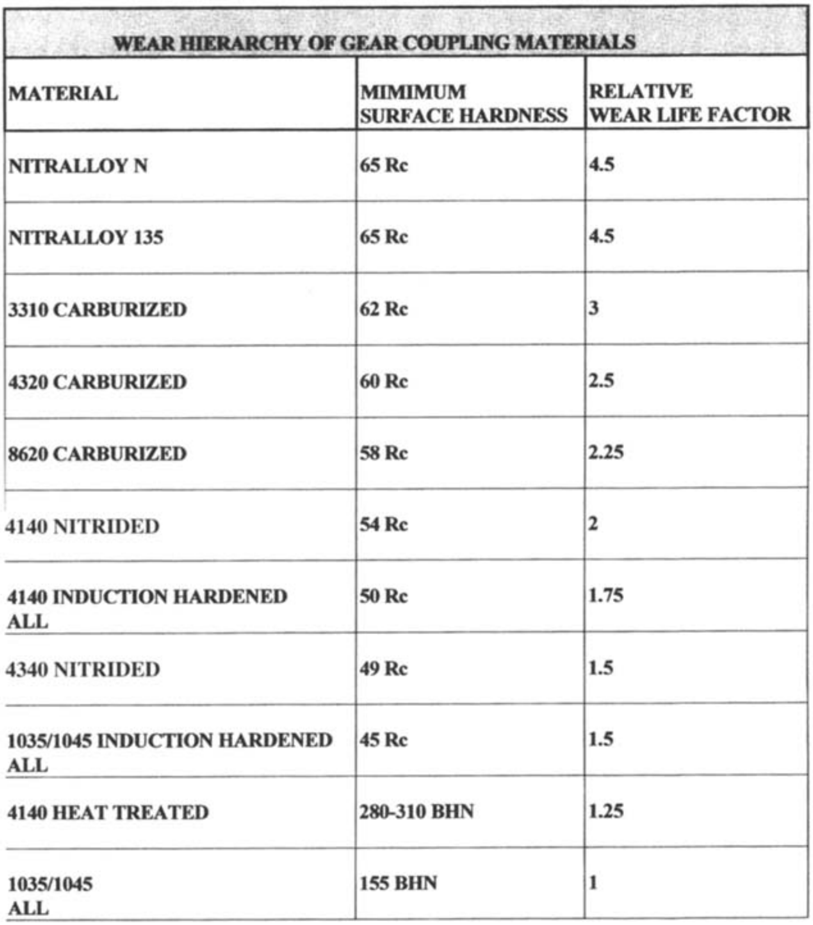
FIGURE 6.43 Various types of heat-treated gear teeth.



STRENGTH HIERARCHY OF GEAR COUPLING MATERIALS

MATERIAL	MIMIMUM HARDNESS	RELATIVE STRENGTH
3310 CARBURIZED	360	5
4320 CARBURIZED	310	4
NITRALLOY N	400	3.5
8620 CARBURIZED	300	3.5
4140 FULLY CONTOURED INDUCTION HARDENED	300	3.25
NITRALLOY 135	320	2.75
4340 NITRITED	330	2.75
4140 FLANK INDUCTION HARDENED	270	2.5
4140 HT AND NITRITED	310	2.5
1035/1045 FULLY CONTOURED INDUCTION HARDENED	155	1.75
1035/1045 INDUCTION HARDENED	155	1
1035/1045	155	1

FIGURE 6.44 Relative strength of various materials used for gear couplings.



WEAR HIERARCHY OF GEAR COUPLING MATERIALS

MATERIAL	MIMIMUM SURFACE HARDNESS	RELATIVE WEAR LIFE FACTOR
NITRALLOY N	65 Rc	4.5
NITRALLOY 135	65 Rc	4.5
3310 CARBURIZED	62 Rc	3
4320 CARBURIZED	60 Rc	2.5
8620 CARBURIZED	58 Rc	2.25
4140 NITRIDED	54 Rc	2
4140 INDUCTION HARDENED ALL	50 Rc	1.75
4340 NITRIDED	49 Rc	1.5
1035/1045 INDUCTION HARDENED ALL	45 Rc	1.5
4140 HEAT TREATED	280-310 BHN	1.25
1035/1045 ALL	155 BHN	1

FIGURE 6.45 Relative wear characteristics of various materials used for gear couplings.

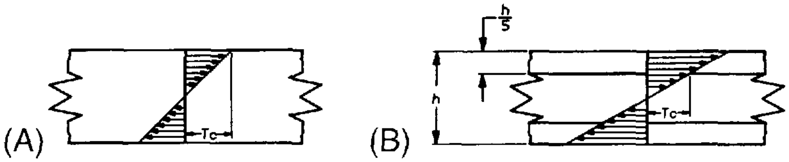


FIGURE 6.46 Stress distribution. (A) Cantilever beam. (B) Proportions to tooth depth.

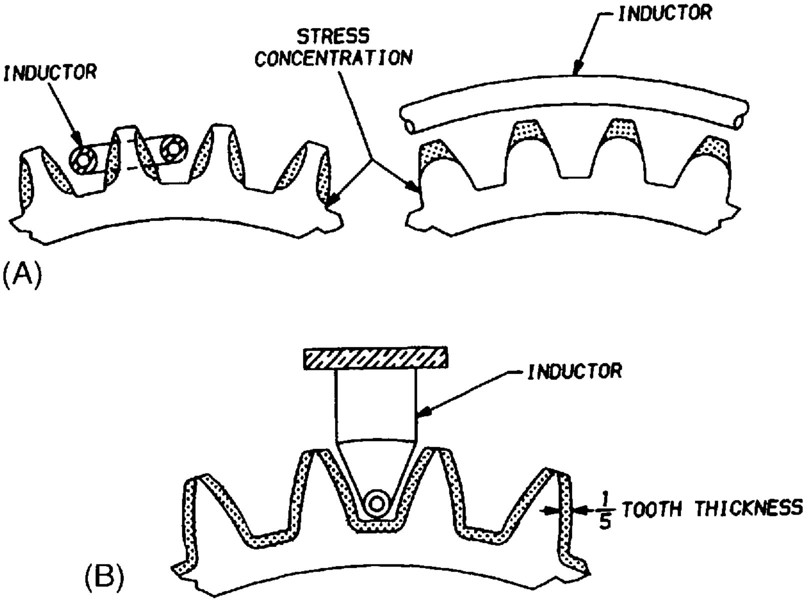


Figure 6.47 Induction hardening. (A) Conventional flank hardening. (B) Fully contoured hardening

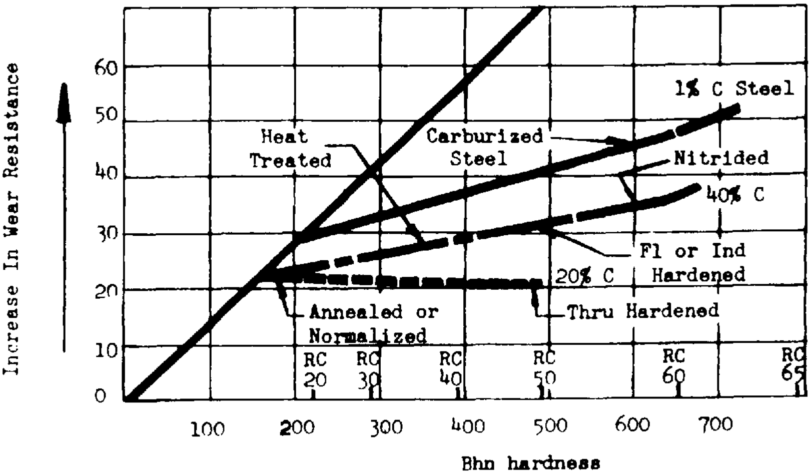


Figure 6.48 Wear resistance versus hardness for various heat treatments.

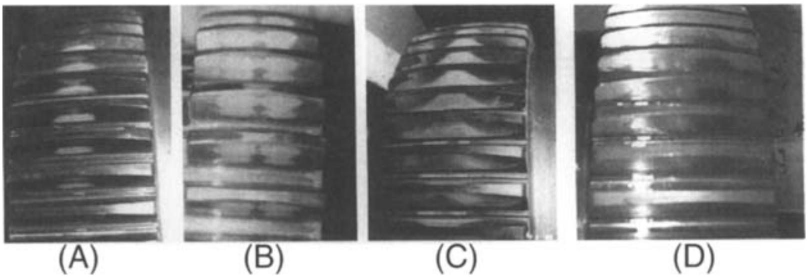


FIGURE 6.49 Blue contact check patterns. (A) As carburized tooth contact pattern. (B) Lapped carburized tooth contact pattern. (C) Ground teeth after carburizing tooth contact pattern. (D) Carburized/ground tooth wear pattern after 1 year operation courtesy of Kop-Flex, Emerson Power Transmission Corp.

C = % in Contact	Misalignment
80%	0°
60%	½°
50%	1°
40%	2°
35°	3°
25°	4°
15%	5°
10%	6°

FIGURE 6.50 Percentage of teeth in contact versus misalignment.

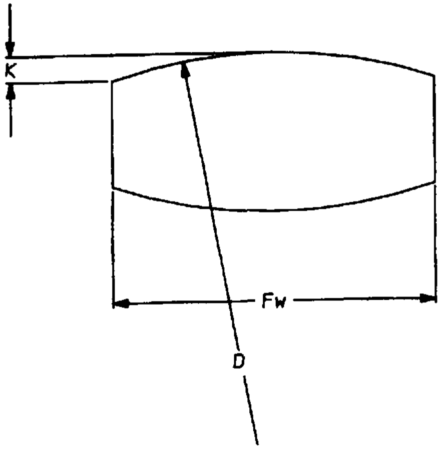


FIGURE 6.51 Amount of tooth crown.

Material	Lubrication Method	S_cV (psi-ips)
Medium Carbon Steel AISI 1032-1045	Grease	125,000-175,000
Alloy Steel (280-310 BHN)	Continuously Lubricated	175,000-225,000
Alloy Steel (280-310 BHN)	Grease Lubricated	200,000-300,000
AISI 4140 (nitrided)	Oil (Continuously lubed)	200,000-300,000
AISI 4140 (nitrided)	Grease lubricated	400,000-600,000
AISI 8620/4320 (Carburized)	Grease	300,000-400,000
AISI 3310 (Carburized)	Grease	400,000-500,000
Nitralloy 135 & N (nitrided)	Grease	1,000,000-1,200,000

FIGURE 6.52 S_cV criteria for gear couplings.

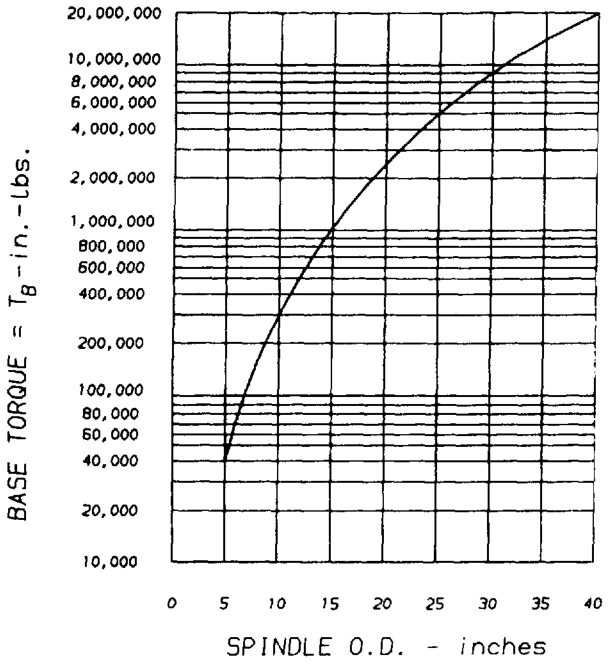


FIGURE 6.53 Torque capacity versus outside diameter.

Maximum Operating Angle	Misalignment Factor (M)
1°	1.0
2°	0.7
3°	0.5
4°	0.4
5°	0.3
6°	0.2

FIGURE 6.54 Misalignment factors.

Size Limit	Material and HT	S = Normal Factor at Maximum Operating Angles		
		<4°	5°	6°
All	1045 flank hardened (flame or induction)	1	1	1
12"—28" 4° maximum	1045 full contoured induction hardened	1.5		
All	4140 flank hardened (flame or induction)	1.5	1.5	1.5
12"—28"	4140 full contoured induction hardened	2.75	2.75	2.50
To 13"	4140 & 4340 nitrided	2.75	2.5	2.25
All	8620, 9310, 3310 carburized	3.5	3.5	3.25

FIGURE 6.55 Material factors.

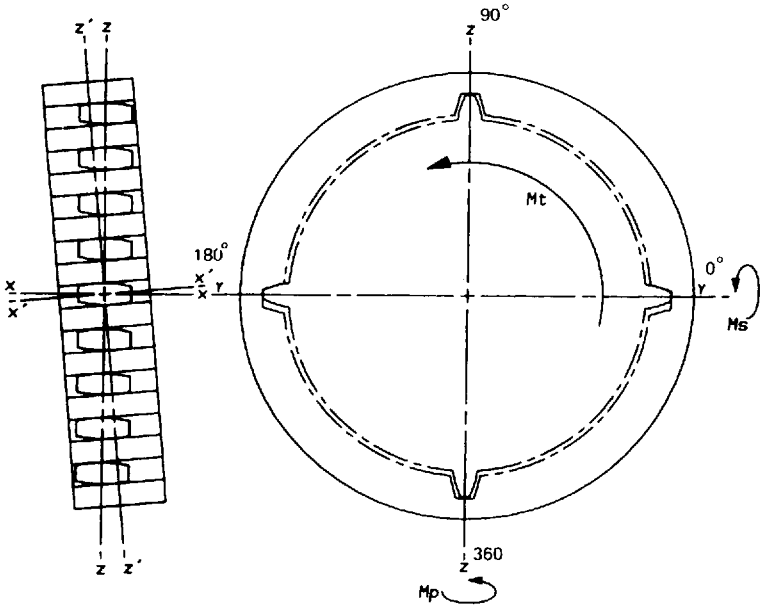


FIGURE 6.56 Coordinate system for a gear coupling

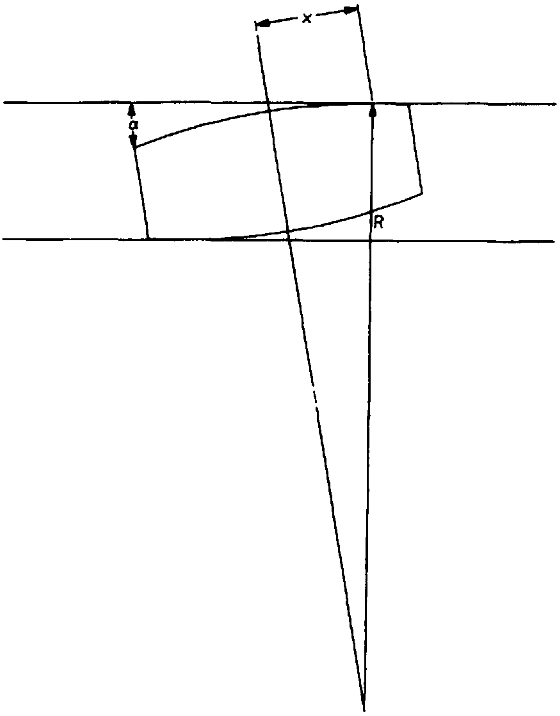


Figure 6.57 Gear tooth sweep.

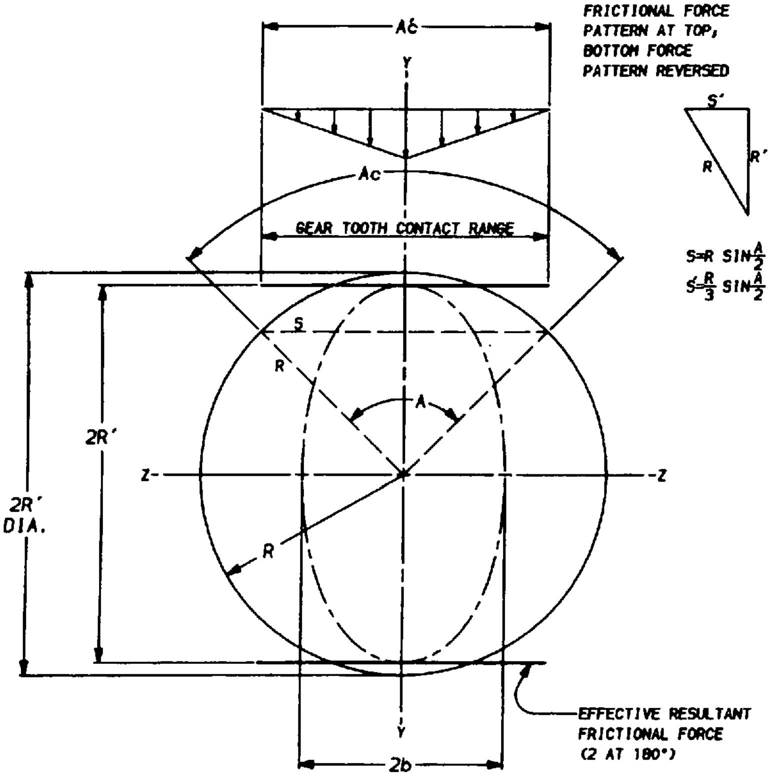

FRICTIONAL FORCE PATTERN AT TOP, BOTTOM FORCE PATTERN REVERSED
Aĉ
Y
Ac
GEAR TOOTH CONTACT RANGE
S'
R
R'
S=R SIN$\frac{A}{2}$
S'=$\frac{R}{3}$ SIN$\frac{A}{2}$
S
R
A
2R'
Z
Z
2R' DIA.
R
EFFECTIVE RESULTANT FRICTIONAL FORCE (2 AT 180°)
Y
2b

Figure 6.58 Average radius for a misaligned gear coupling.

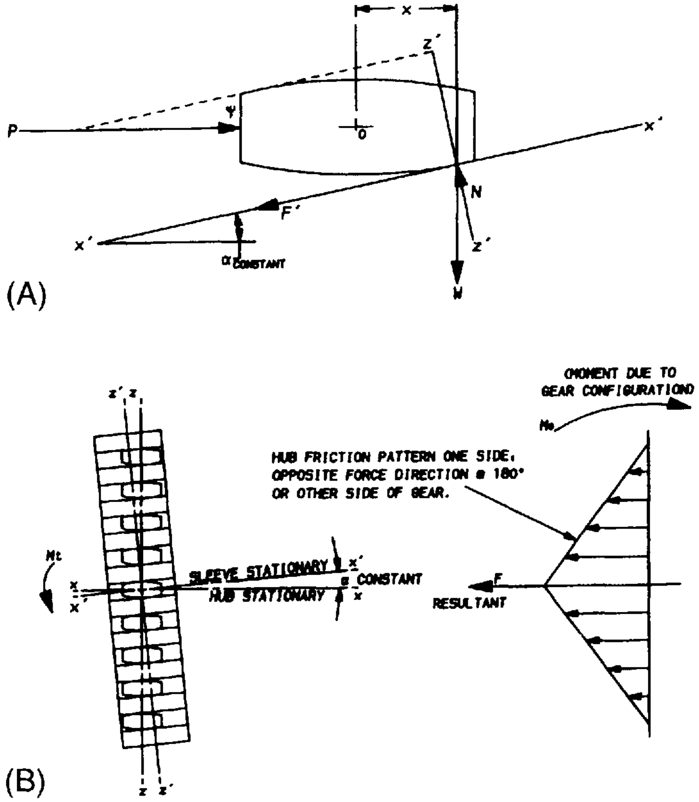

x
z'
Ψ
P
o
x'
N
F'
x'
z'
α CONSTANT
W
(A)
(MOMENT DUE TO GEAR CONFIGURATION)
Mo
z' z
HUB FRICTION PATTERN ONE SIDE, OPPOSITE FORCE DIRECTION @ 180° OR OTHER SIDE OF GEAR.
Mt
SLEEVE STATIONARY
x'
α CONSTANT
HUB STATIONARY
x
x
x'
F
RESULTANT
(B)
z z'

Figure 6.59 Gear tooth loads. (A) Forces in a gear coupling. (B) Moment pattern in a gear coupling.

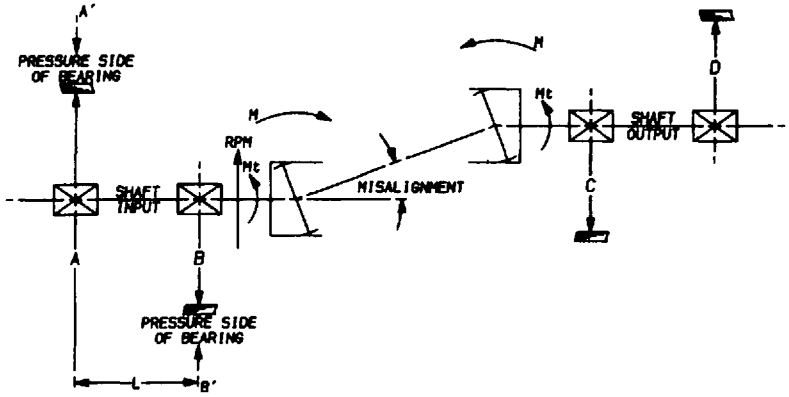


Figure 6.60 Plan view of bearing reactions.

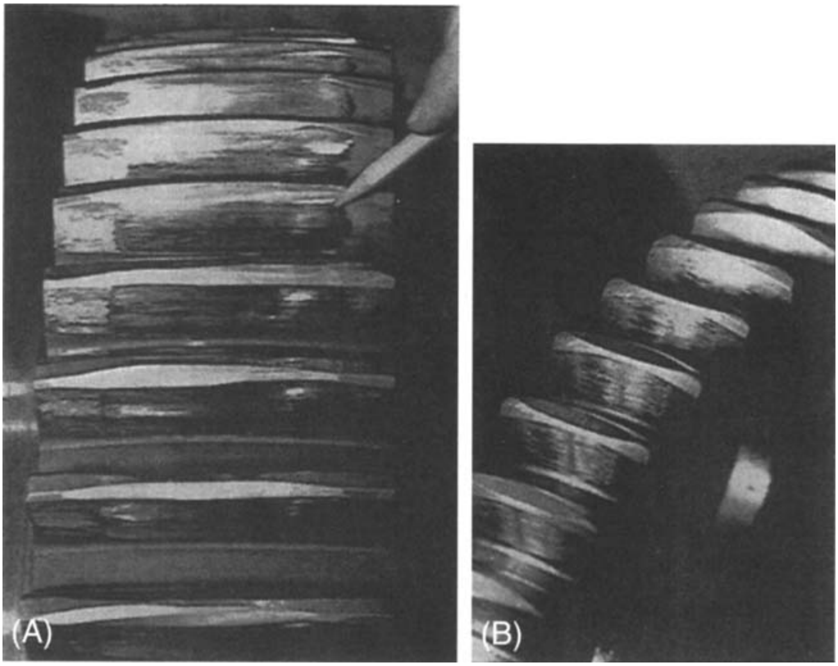


FIGURE 6.61 Typical good tooth wear patterns. (A) Gear spindle. (B) Aircraft gear coupling. (Courtesy of Ameridrives International.)

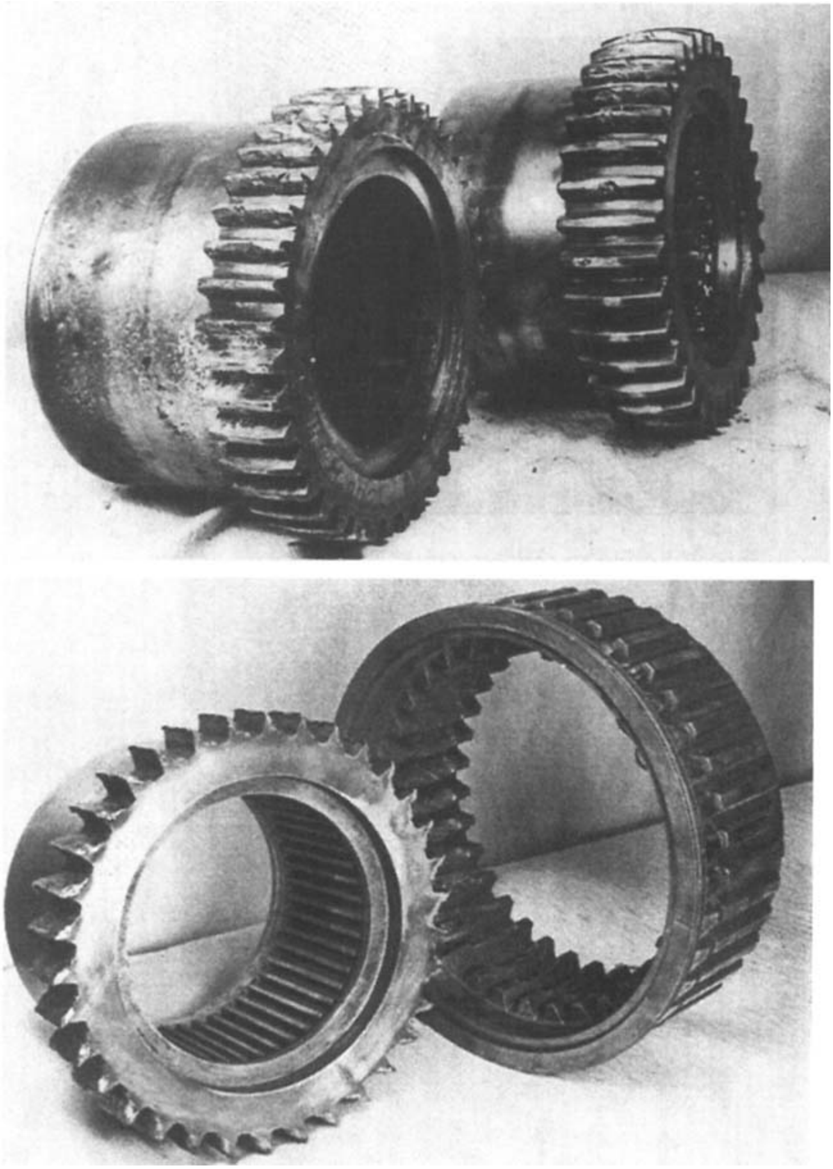


Figure 6.62 Failure due to lubricant breakdown.

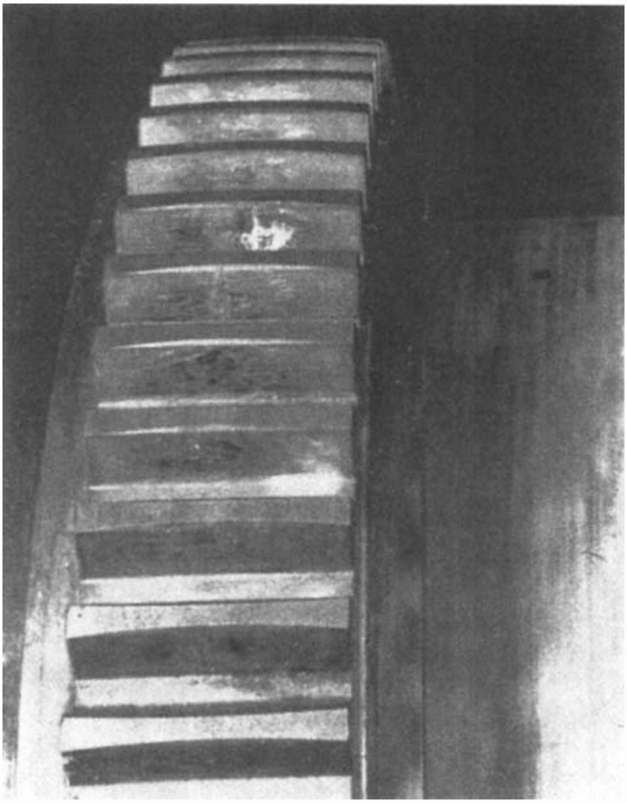


FIGURE 6.63 Improper tooth contact.

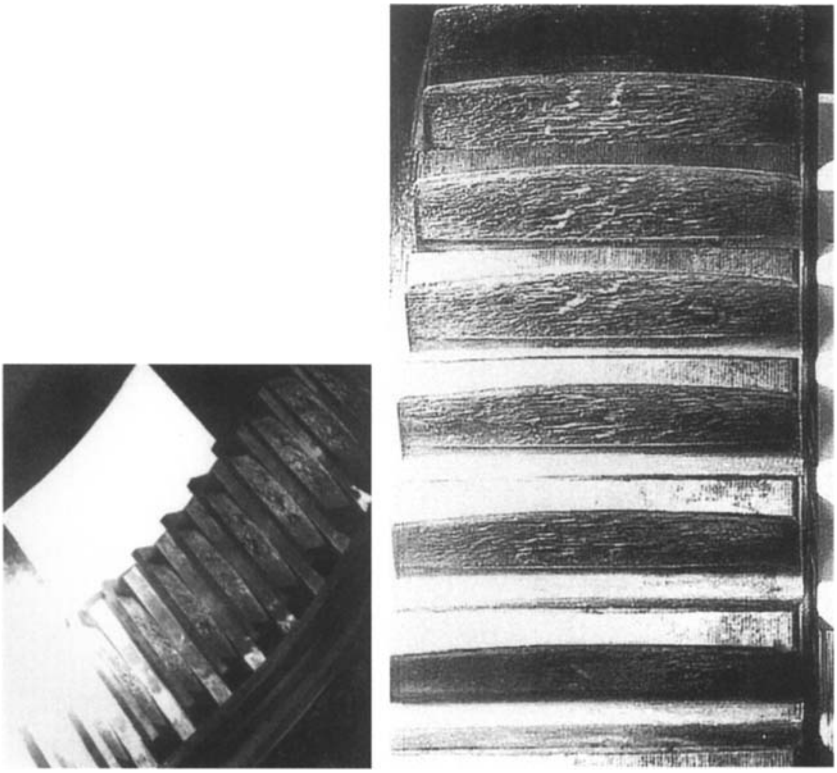


Figure 6.64 Lack of lubrication. (*Left*) Sleeve tooth. (*Right*) Hub tooth.

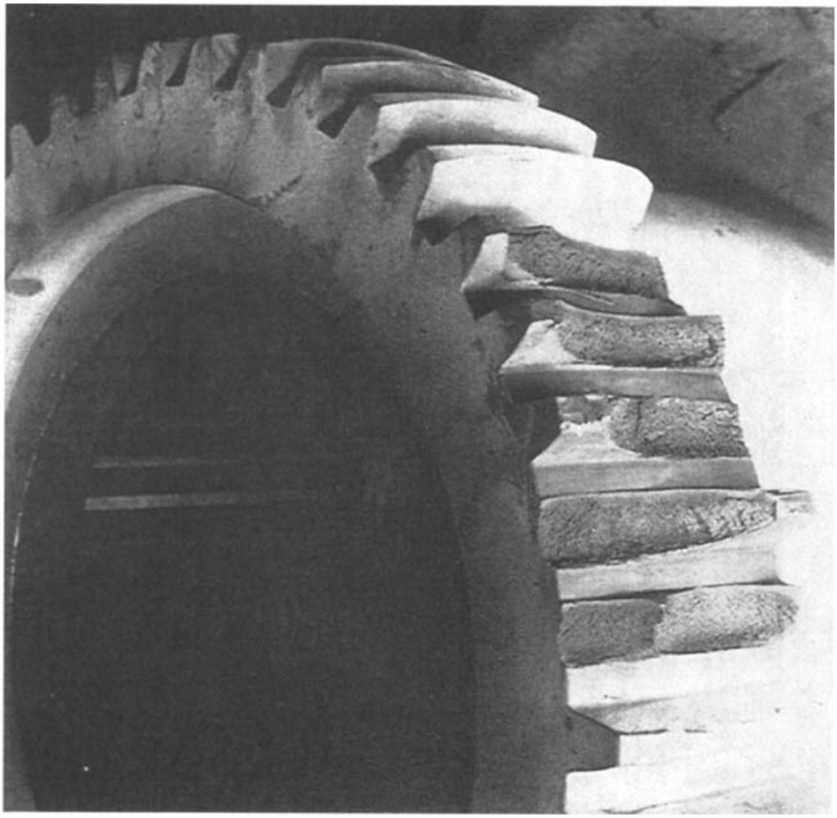


FIGURE 6.65 Failure due to overmisalignment.

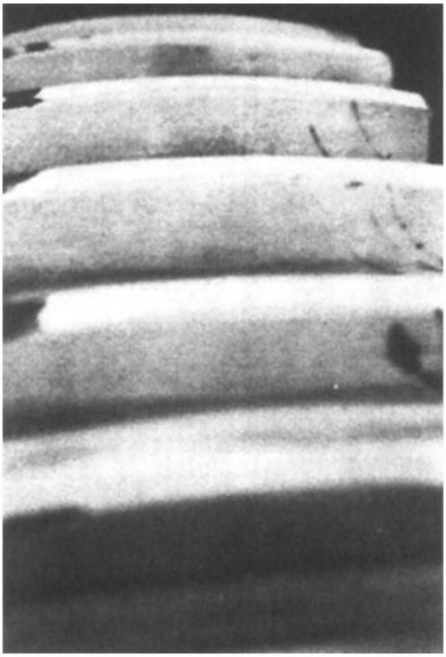


FIGURE 6.66 Failure due to overmisalignment.

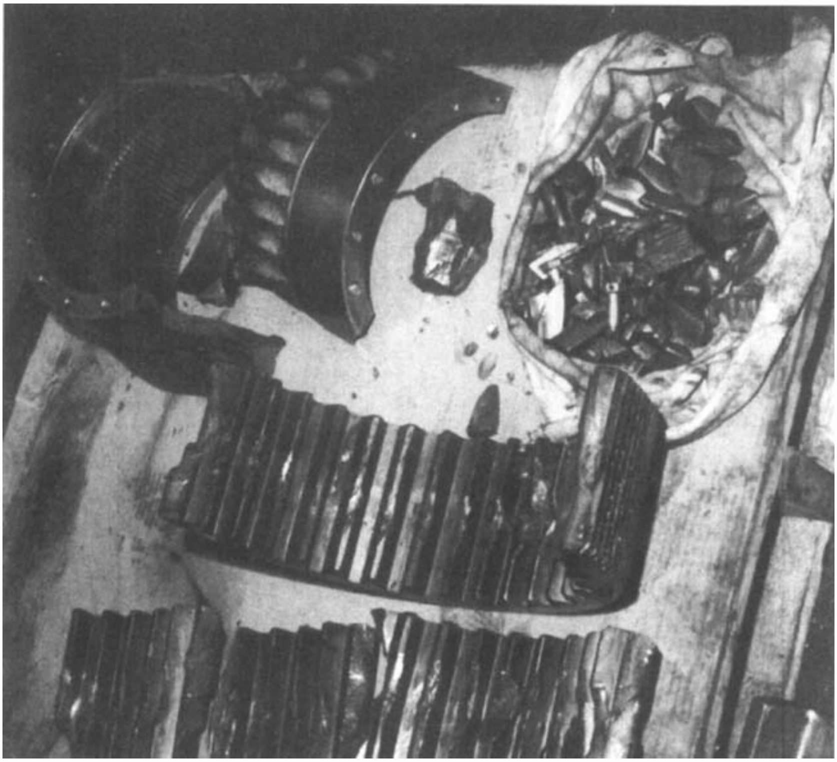


Figure 6.67 Failure due to static overmisalignment.

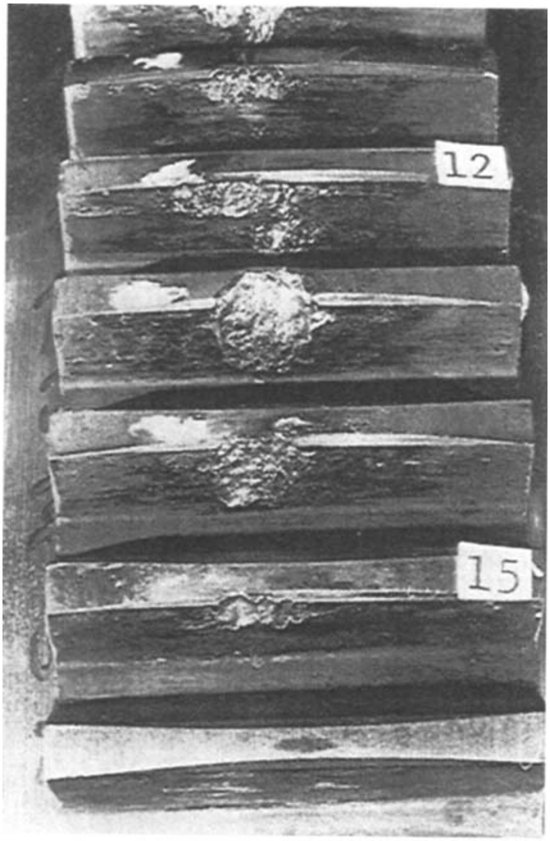


FIGURE 6.68 Failure due to operating at a torsional critical speed.

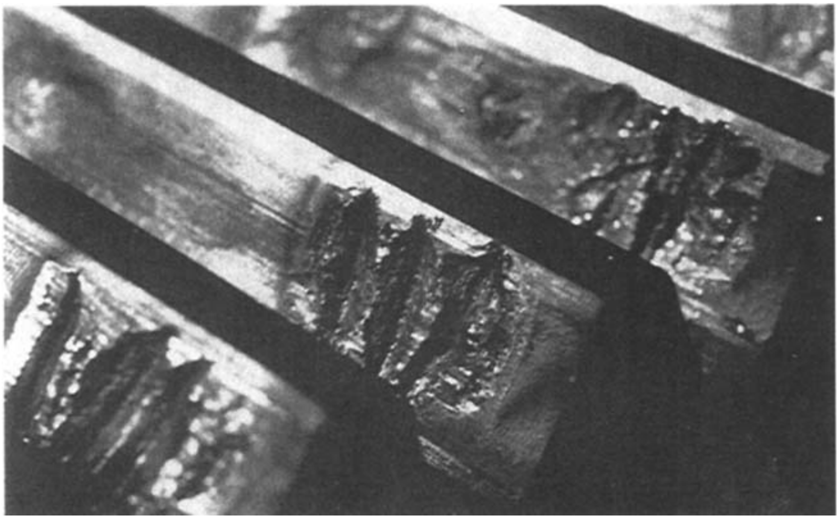


FIGURE 6.69 Worm tracking failure.

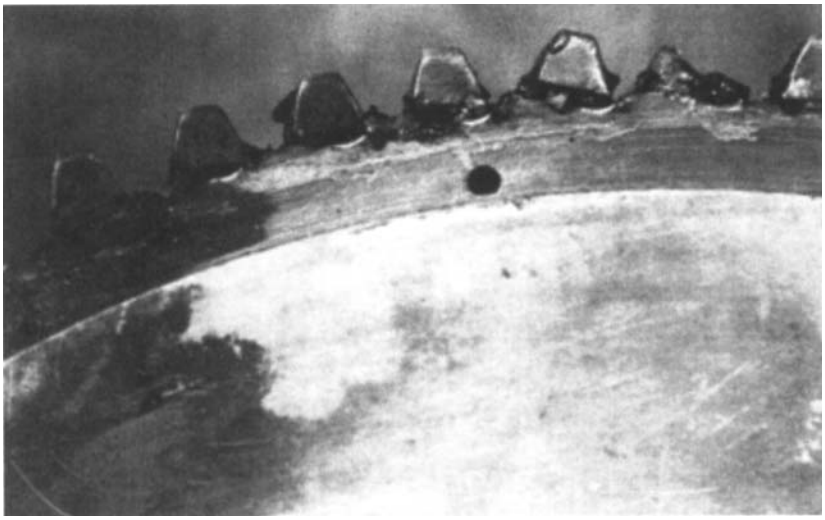


FIGURE 6.70 Sheared gear tooth.

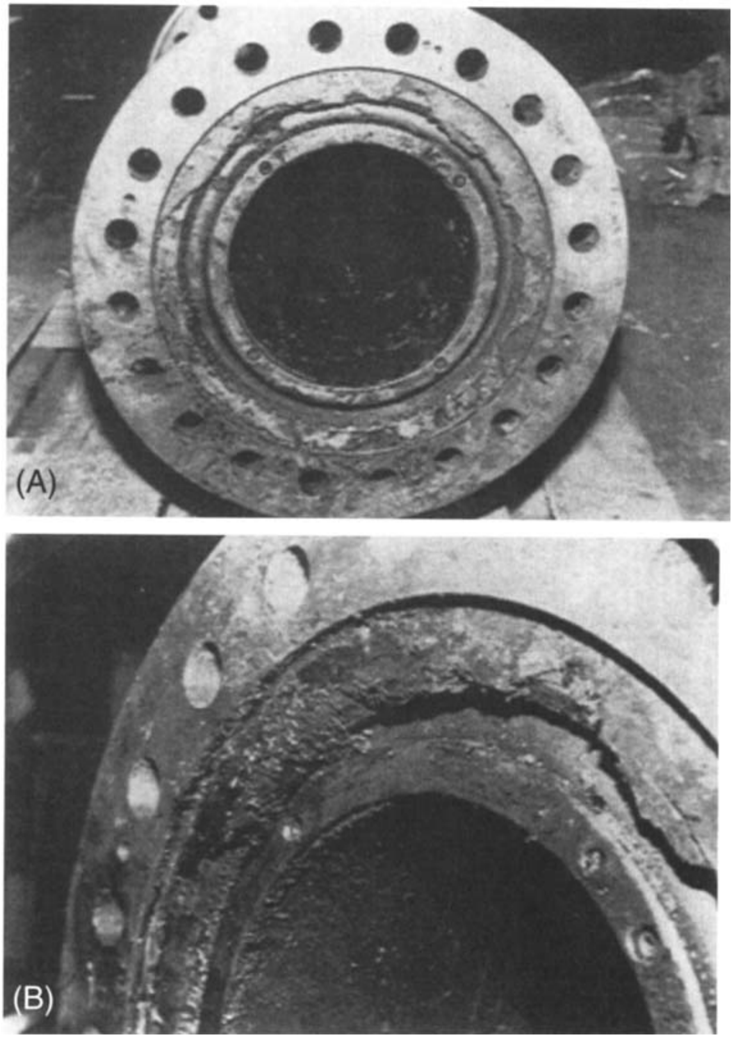


Figure 6.71 Sludge-contaminated gear coupling.

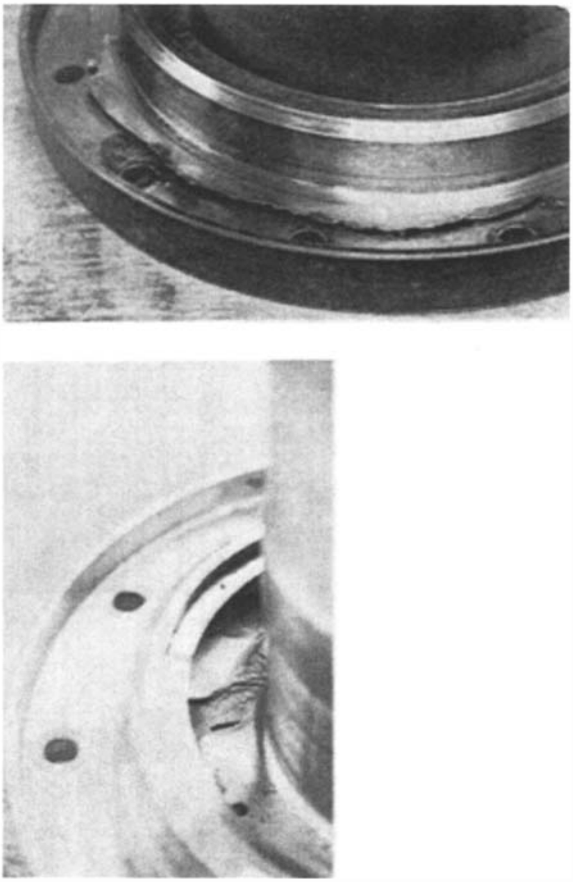


FIGURE 6.72 Broken spacer from locked-up gear teeth.

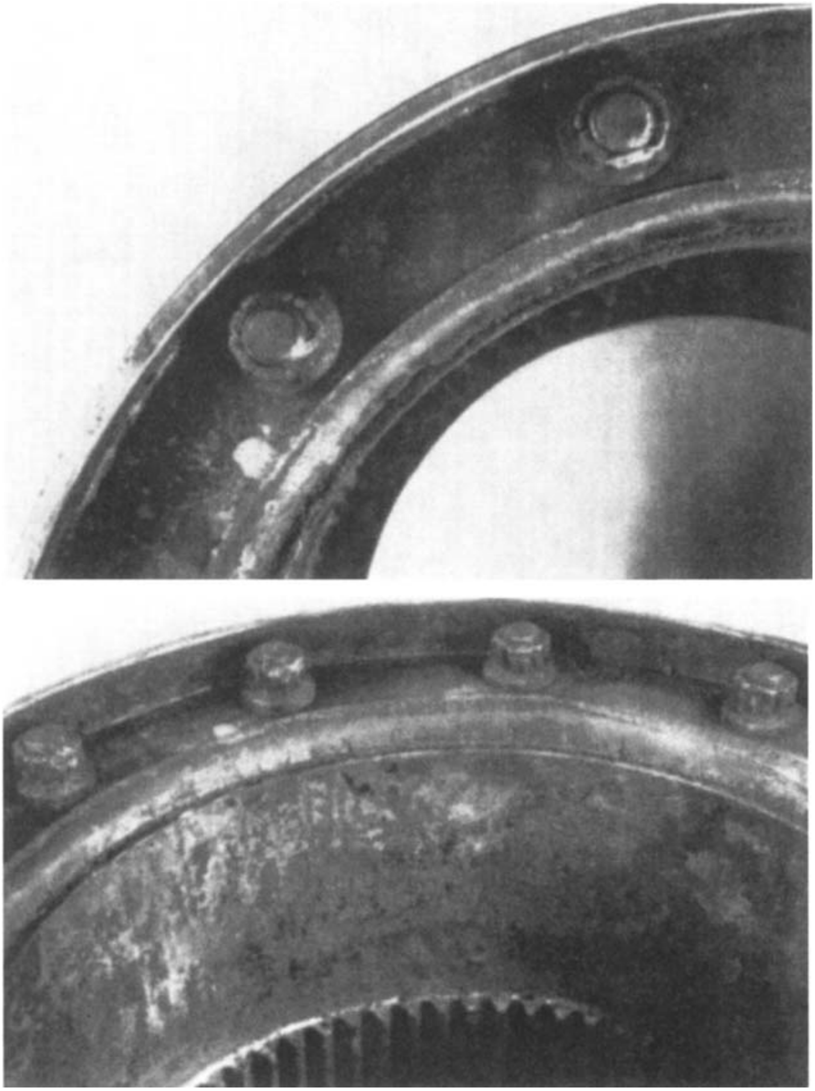


FIGURE 6.73 Corroded gear coupling components.

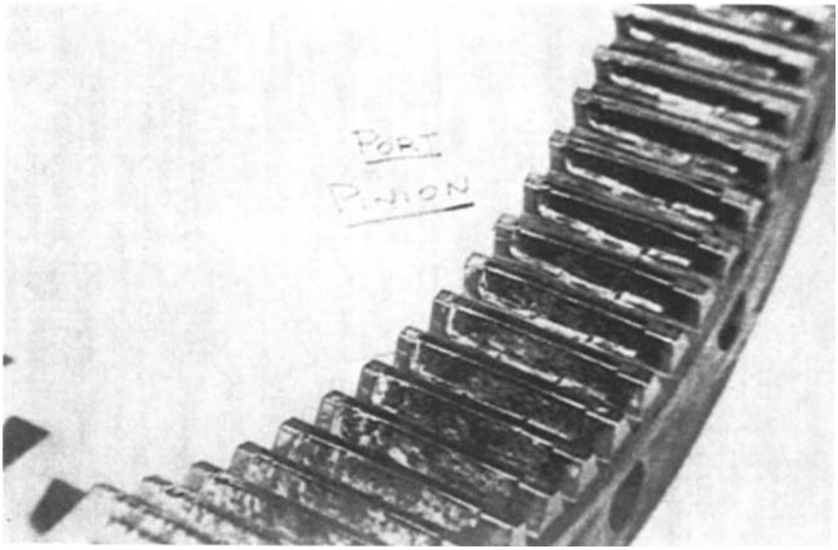


FIGURE 6.74 Mechanical tooth lock-up from embedding. (From Michael Neale et al., Proceedings of the International Conference on Flexible Coupling for High Power and Speed, 1977.)

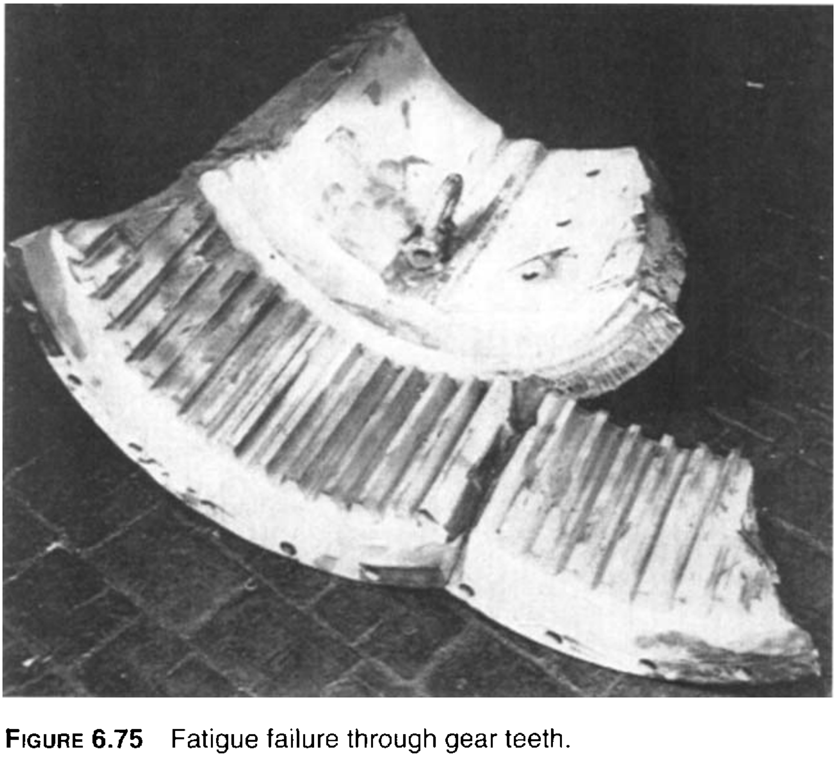


FIGURE 6.75 Fatigue failure through gear teeth.

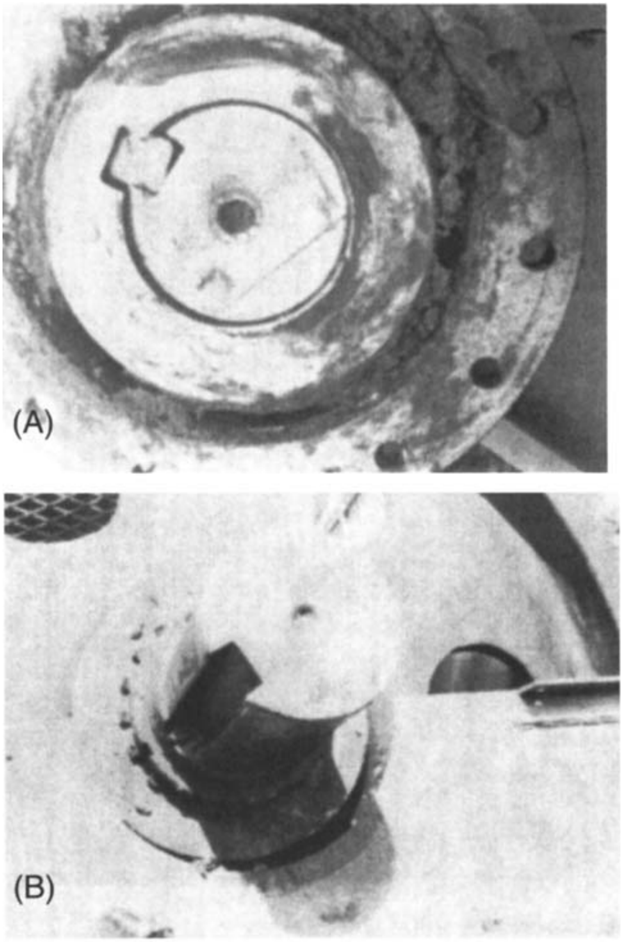


FIGURE 6.76 Key shear rollover.

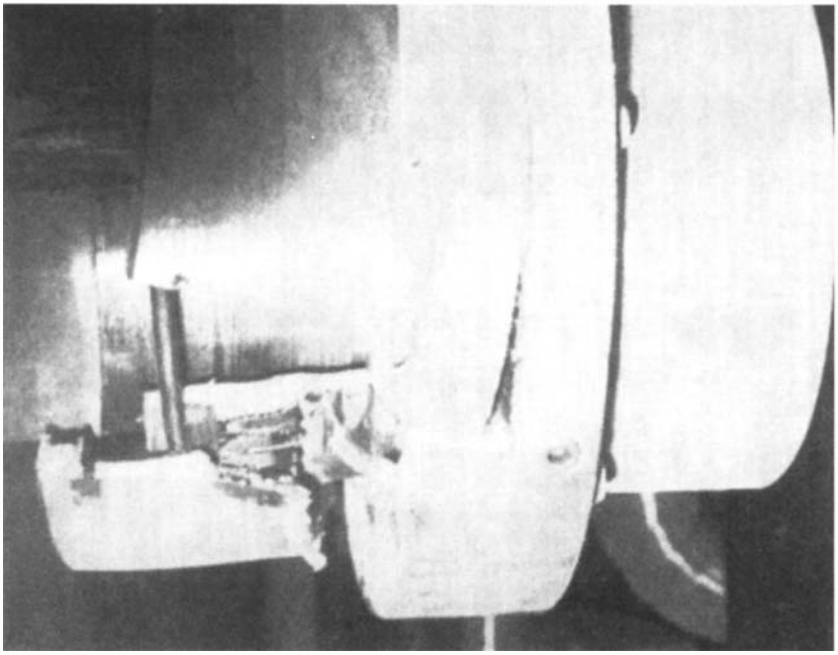


FIGURE 6.77 Hub bursting failure.

III. GRID COUPLINGS*

The basic metallic grid coupling (see Figure 6.78) consists of two hubs that contain slots, which may be straight, tapered, or contoured. The two hubs are connected by a "serpent"-style metallic grid (spring).

Metallic grid couplings have been used successfully for more than 60 years. Figure 6.79 shows how a grid coupling accommodates for the following three types of misalignment:

1. *Parallel misalignment:* Movement of the gird in the lubricated grooves accommodates for parallel misalignment (Figure 6.79A).
2. *Angular misalignment* (Figure 6.79B): The grid–groove design permits a rocking and sliding action of the lubricated grid and hubs.
3. *Axial movement* (Figure 6.79C): Unrestrained end float for both driving and driven members is permitted because the grid slides in the lubricated grooves. The amount of axial movement can also be limited.

A. Torsional Flexibility of Metallic Grid Couplings

Torsional flexibility is the unique ability of a grid coupling to torsionally deflect when subjected to normal, shock, or vibratory loads, providing flexible accommodation to changing load conditions. Consequently, the coupling is capable of damping vibration and of reducing peak or shock loads.

The following description shows how this is accomplished. Under light loads the grid bears near the outer edges of the hub teeth (Figure 6.80A). The long span between the points of contact remains free to flex under load variations. Under normal loads, as the load increases, the distance between the supports on the hub teeth is shortened, but a free span remains to cushion shock loads (Figure 6.80B). Under shock load, the coupling is flexible within its rated capacity (Figure 6.80C). Under extreme overloads, the grid bears fully on the hub teeth and transmits the full load directly.

---

*Figures for this section are grouped together starting on p. 347.

B. Where Metallic Grid Couplings Are Used

The metallic grid coupling has been applied to nearly all types of industrial applications, from centrifugal pumps to steel mill applications. Some typical applications are:

Agitators
Blowers
Compressors
Cranes
Elevators
Fans
Generators
Machine tools
Mixers
Paper mills
Pumps
Rubber industry
Steel mills and auxiliary equipment
Textile industry

There are sizes to satisfy a 1-hp motor with a 7/8-in. bore or for a BOF (basic oxygen furnace) vessel tilting shaft 40 in. in diameter. Several applications of the grid coupling are as follows:

Friction roll drives (Figure 6.81)
Conveyor drives (Figure 6.82)
Centrifugal pumps (Figure 6.83)

C. Types of Metallic Grid Couplings

There are two basic types of grid couplings:

The horizontal split covered grid coupling (Figure 6.84A)
The vertical split covered grid coupling (Figure 6.84B)

A third common type of grid coupling is the spacer type (Figure 6.85). The spacer-type grid coupling is especially suited for pump applications.

The grid coupling is also commonly used with:

A brake drum or wheel (Figure 6.86A)
A friction disk (Figure 6.86B)
A limited end float for use on motors (Figure 6.87)

D. Design and Construction of Metallic Grid Couplings

1. Metallic Grid Coupling Ratings

Ratings are usually based on grid fatigue strength. Because of the various grid forms and slots, the design is based primarily on empirical data accumulated over the years. The grids are usually made of spring steel—either high carbon steel or chrome–vanadium steel. In evaluating grid couplings, it is best to ask coupling manufacturers for the following information:

1. Grid material
2. Grid material properties
   a. Yield strength
   b. Endurance limit
3. Stresses in grid for:
   a. Normal torque, no misalignment
   b. No torque, full angular misalignment
   c. No torque, full axial travel

2. Construction

The hubs of a grid coupling are usually made of medium carbon steel. Without the following information, bore capacities cannot be compared:

1. Hub material
2. Hub properties
3. Ultimate strength
4. Yield strength
5. Endurance limit
6. Hub OD and length dimensions

E. Selection of Metallic Grid Couplings

See Figure 6.88 for typical selection procedure.

F. How Metallic Grid Couplings Affect a System

A grid coupling has two very significant effects on a system. It reduces peak shock or impulsive loads by softening (low torsional stiffness) the elastic system, and it absorbs or dampens energy. The torque developed when one is trying to accelerate a two-mass system is determined using the following formula:

$$T_m = \frac{I_1 I_2}{I_1 + I_2} (\Delta\omega)(\omega_0) \tag{6.31}$$

where

T_m = maximum torque
I_1 = mass moment of inertia-driven equipment
I_2 = mass moment of inertia of the driver
ω_0 = natural frequency of system
$\Delta\omega$ = change of velocity

$$\omega_0 = \frac{K_t(I_1 + I_2)}{I_1 I_2} \tag{6.32}$$

where K_t represents the stiffness of the total system and Is determined by the formula

$$K_t = \frac{1}{1/K_S + 1/K_C} \tag{6.33}$$

where

K_S = stiffness of system
K_C = stiffness of coupling

From the preceding calculations it can be seen that the softer the coupling, the lower the maximum torque. The resilience or damping of the coupling will tend to reduce the peak torque markedly, as shown In Figure 6.89.

G. Failure Mode for Metallic Grid Couplings

Two of the most common types of failure are wear, usually caused by improper, lost, or deteriorated lubricant, and fatigue of the grids, usually caused by overmisalignment. Other types of failures that have occurred include:

1. *Hub bursting:* results from overtorque or from an improper key/keyway fit
2. *Key shear:* usually caused by overload
3. *Seal failure:* usually caused by loss of lubricant, resulting in excessive heat and deterioration of the seal
4. *Breakage, bolts, spacers, etc.:* typically caused by abuse, overtorqued bolts, overmisalignment, or poor installation and maintenance practices

In general, most coupling failure occurs as a result of misalignment, misuse, abuse, or neglect rather than resulting from a design or material deficiency.

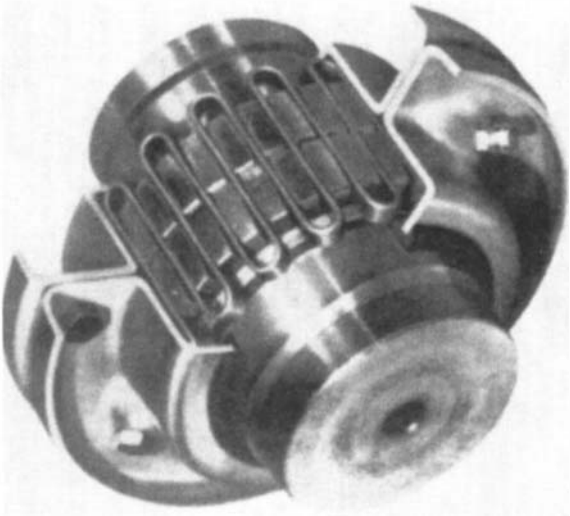


FIGURE 6.78 Basic metallic grid coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

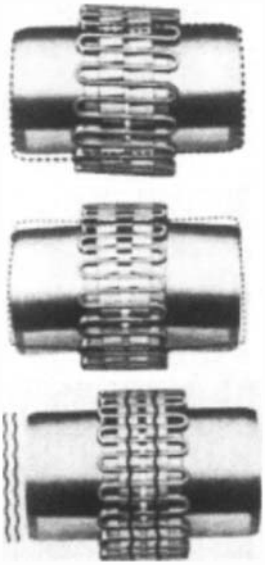


FIGURE 6.79 How a grid coupling accommodates misalignment. (Courtesy of Falk Corporation.)

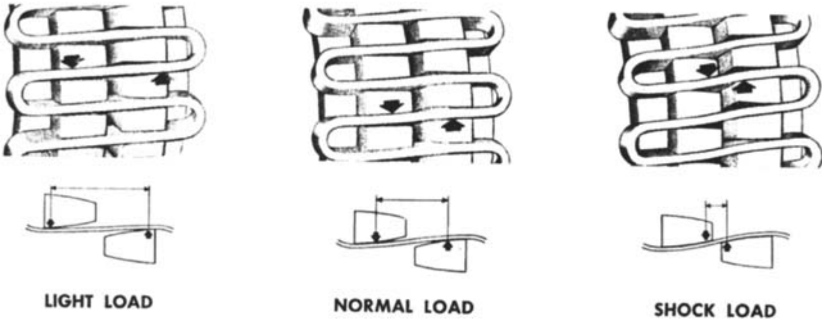


Figure 6.80 How a grid coupling accommodates torsional flexibility. (Courtesy of Falk Corporation.)

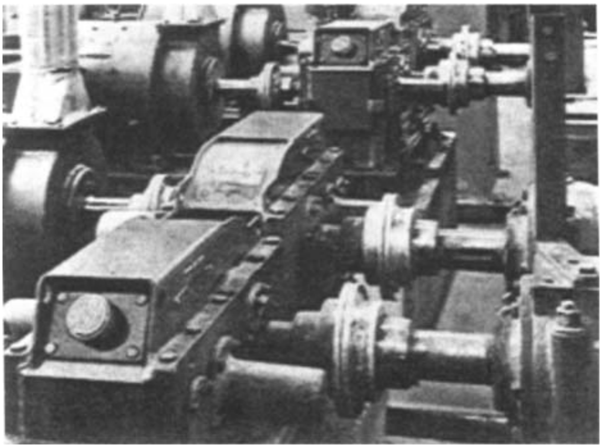


Figure 6.81 Grid coupling for friction roll drive. (Courtesy of Falk Corporation.)

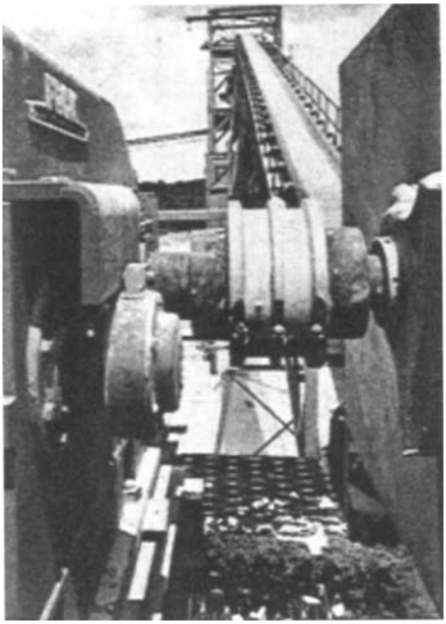


FIGURE 6.82 Grid coupling for conveyer drive. (Courtesy of Falk Corporation.)

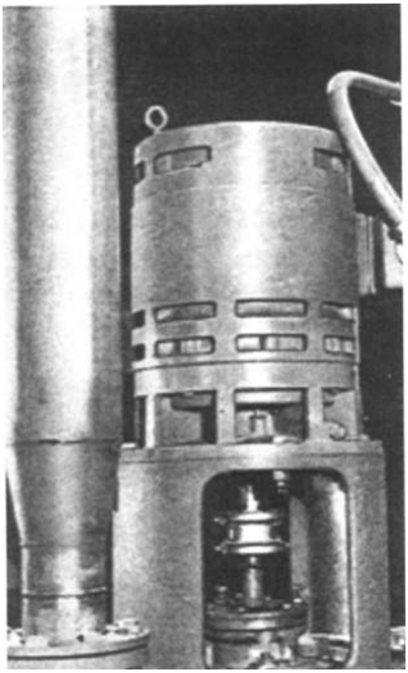


FIGURE 6.83 Grid coupling for centrifugal pump. (Courtesy of Falk Corporation.)

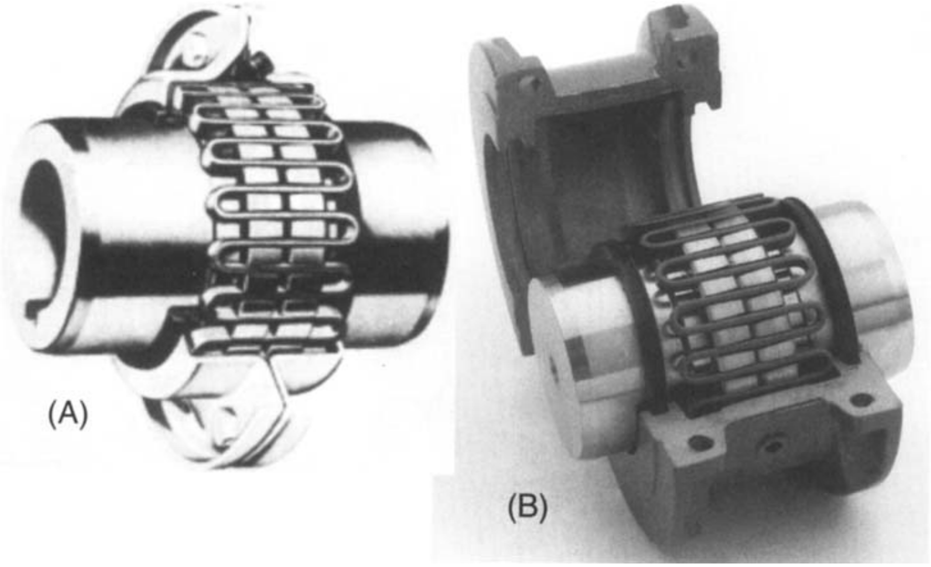


FIGURE 6.84 Covered grid couplings. (A) Vertical split covered grid coupling (courtesy of Falk Corporation). (B) Horizontal split covered grid coupling (courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation).

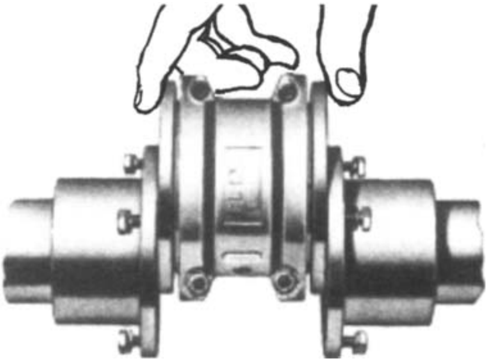


FIGURE 6.85 Spacer-type grid coupling. (Courtesy of Falk Corporation.)

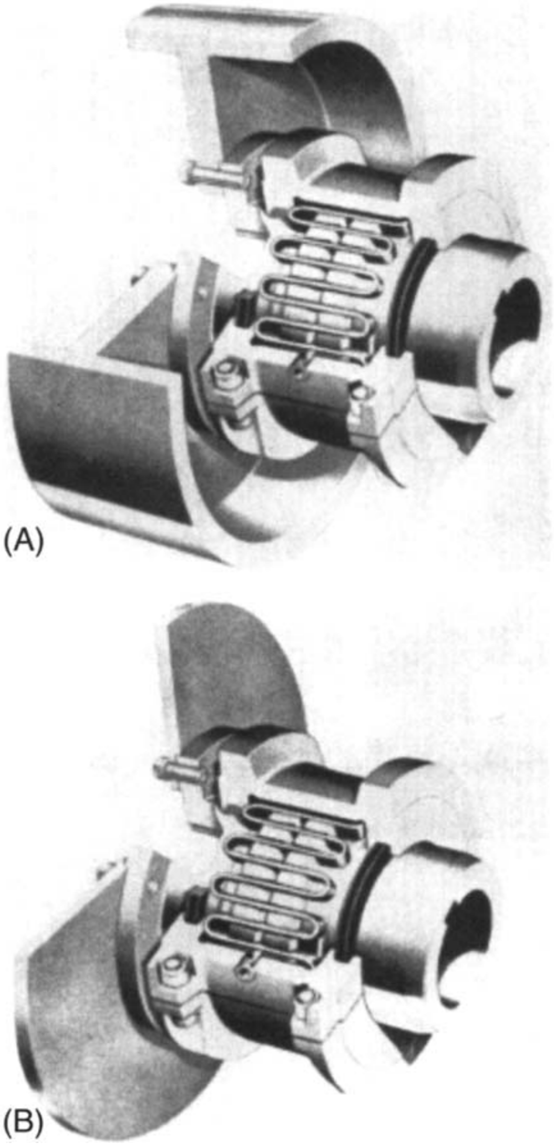


FIGURE 6.86 (A) Brake drum grid coupling. (B) Friction disk grid coupling. (Courtesy of Falk Corporation.)

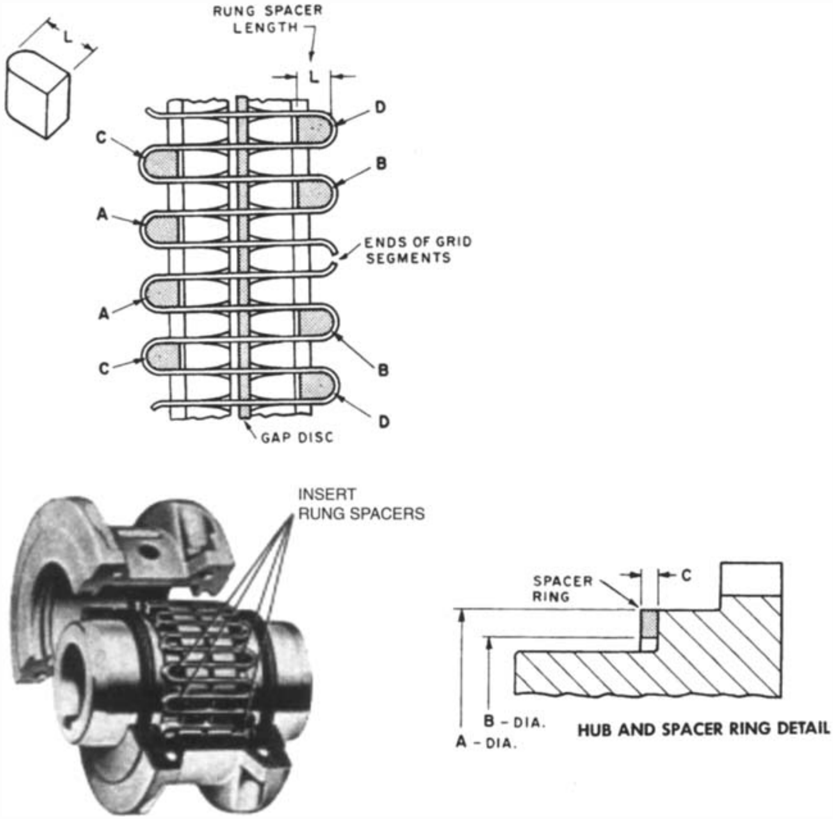


FIGURE 6.87 Limited end float grid coupling (Courtesy of Falk Corporation.)

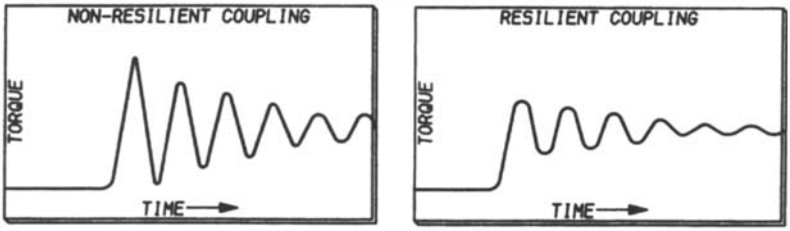


FIGURE 6.89 Resilient damping of a grid coupling.

I. Coupling Type:
Select the appropriate KOP-GRID coupling type for your application. See page 1 for coupling types.

II. Coupling Size:
Step 1: Determine the proper service factor from pages 10 and 11.

Step 2: Calculate the required HP/100 RPM, using the HP rating of the drive and the coupling speed (RPM) as shown below:

$$\frac{\text{HP} \times \text{SERVICE FACTOR} \times 100}{\text{RPM}} = \text{HP}/100\ \text{RPM}$$

Step 3: Using table below, select the coupling size having a rating sufficient to handle the required HP/100 RPM at the appropriate service factor.

Coupling Size	1020	1030	1040	1050	1060	1070	1080	1090	1100	1110	1120	1130	1140
Coupling Rating *HP/100 RPM*	*0.68*	*1.93*	*3.22*	*5.63*	*8.85*	*13*	*27*	*48*	*81*	*121*	*177*	*257*	*370*

Step 4: Verify that the actual coupling speed (RPM) is equal to or less than the maximum allowable speed rating of the coupling (see pages 4-8).

Step 5: Verify that the maximum bore of the coupling selected (see pages 4-8) is equal to or larger than either of the equipment shafts.

Step 6: Check the overall dimensions to ensure coupling will not interfere with the coupling guard, piping, or the equipment housings and that it will fit the required shaft separation. See pages 4 thru 8 for coupling dimensional data.

How to Order

I. Coupling Size:
Specify size and type. If in doubt, provide the following:

A. Horsepower (HP) to be transmitted by coupling, normal and maximum.
B. Coupling Speed (RPM).
C. Type of Driving machine (motor, engine, etc.).
D. Type of Driven machine or equipment.
E. Driving and Driven shaft diameters.
F. Shaft separation (for spacer style application).

II. Bore & Keyway:
Specify bore and keyway dimension for both ends.

A. **Clearance Fit:**
   Sizes 1020 through 1090 are supplied with one set screw over the keyway, for clearance fit, unless otherwise specified.
B. **Interference Fit:**
   Sizes 1100 through 1140 when bored for interference fit, are not supplied with a set screw, unless otherwise specified.
C. **Tapered Bushing:**
   Specify bushing type and part number for the boring. Kop-Flex does not supply tapered bushings.

FIGURE 6.88 Selection procedure. (Page numbers refer to the original Kop-Flex manual.) (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

IV. CHAIN COUPLINGS

The basic chain coupling consists of two hubs and a length of double-width roller chain (see Figure 6.90). The two hubs have sprockets and are connected by a length of roller chain. Misalignment is compensated for by the clearances between the chain and the sprocket (see Figure 6.91)

These couplings have built-in clearances between (1) the chain rollers and the sprocket teeth, (2) the chain rollers and the hardened bushings, (3) the bushings and the pins, and (4) the chain links and the sprocket teeth. All this clearance helps to compensate for angular and parallel offset misalignment as well as shaft end float. Chain couplings can handle up to 2° of angular misalignment and approximately 0.003 in. per inch of coupling outside diameter in offset capacity.

Chain couplings are used on the following types of equipment:

Agitators
Conveyors
Pumps
Feeders
Hoists
Mixers
Machine tools

See Figures 6.92 and 6.93 for typical applications.

A. Types of Chain Couplings

There are three basic types of chain couplings: (1) the double roller chain coupling (Figure 6.94A), (2) the silent chain coupling (Figure 6.94B), and (3) the plastic chain coupling (Figure 6.94C):

1. *The double roller chain type:* The double roller chain type is the most common. It is normally used on applications where speeds are moderate, loads are steady, high static or starting loads are present, and ease of connecting driver and driven units is required.
2. *The silent chain coupling:* The silent chain coupling is used for heavy-duty applications such as on steel mill drives. The high torque capacity is accomplished because this design has a much better load distribution from the chain to the sprocket

---

*Figures for this section are grouped together starting on p. 356.

due to the stacking up of individual links and their angle of engagement with the sprocket teeth.

3. *The plastic chain coupling:* The plastic chain coupling provides a nonlubricating coupling that can be used in the food-processing and textile industries. It is corrosion-resistant and pollution-free and it runs more quietly than metal couplings.

B. Design and Construction of Chain Couplings

Chain couplings are usually constructed of high-tensile-strength, heat-treated roller chain and sprockets. The coupling ratings are usually based on the weakest link in the chain, usually the connecting link. As with most mechanical flexible couplings, the most important design consideration is wear and therefore lubrication. It is considered good practice to enclose and lubricate chain couplings, because this will help to maintain lubrication between the chain and the sprocket and thus reduce the coefficient of friction, thereby increasing the life of the chain and sprocket and reducing the load on the equipment. See Figure 6.95 for typical coupling covers. As a general rule, chain couplings should incorporate covers under the following conditions:

When speeds exceed 500 rpm
In dusty atmospheres
In moist applications

C. Selection

See Figure 6.96 for typical size, torque, and speed limits. See Figure 6.97 for typical selection procedures.

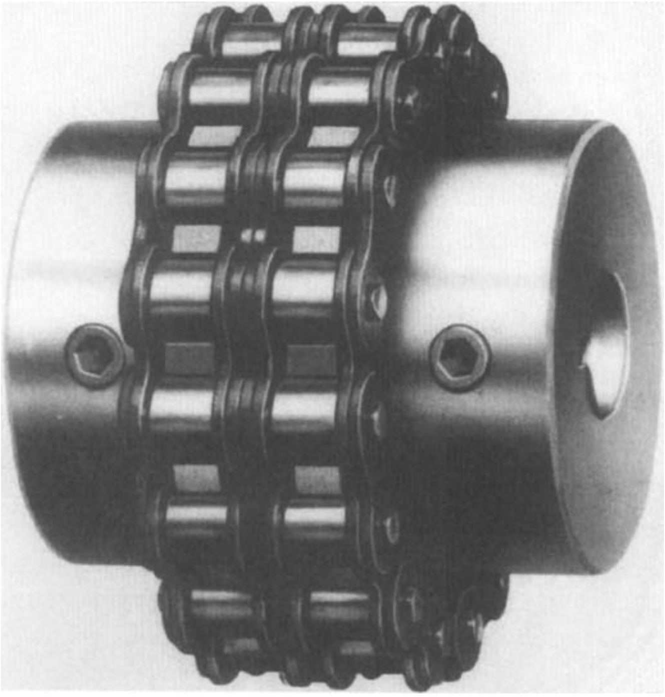


Figure 6.90 Basic chain coupling. (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

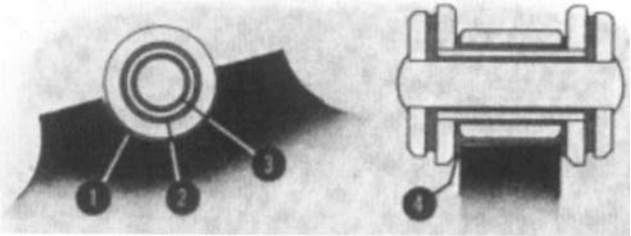


Figure 6.91 Built-in clearances between (1) chain rollers and sprocket teeth, (2) chain rollers and bushings, (3) bushings and pins, and (4) link plates and sprocket teeth.

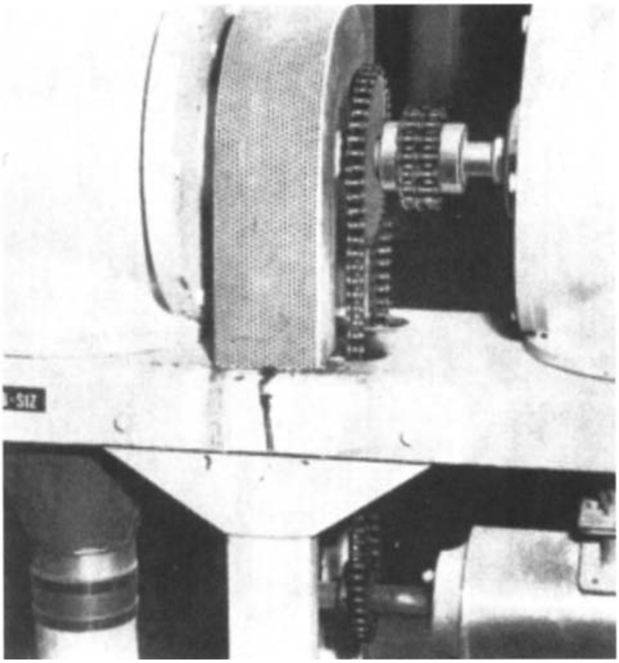


FIGURE 6.92 Grain separator drive. (Courtesy of Dodge Division of Reliance Electric.)

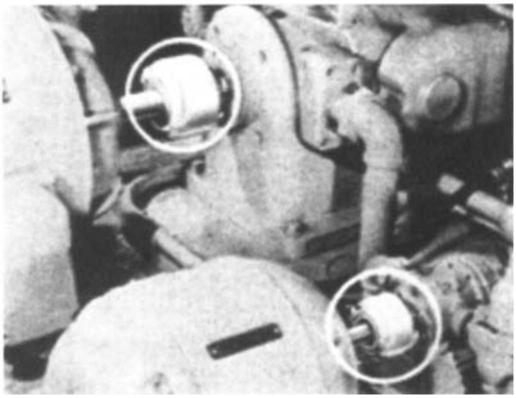


FIGURE 6.93 Chain coupling on a hydraulic power unit (Courtesy of Link-Belt.)

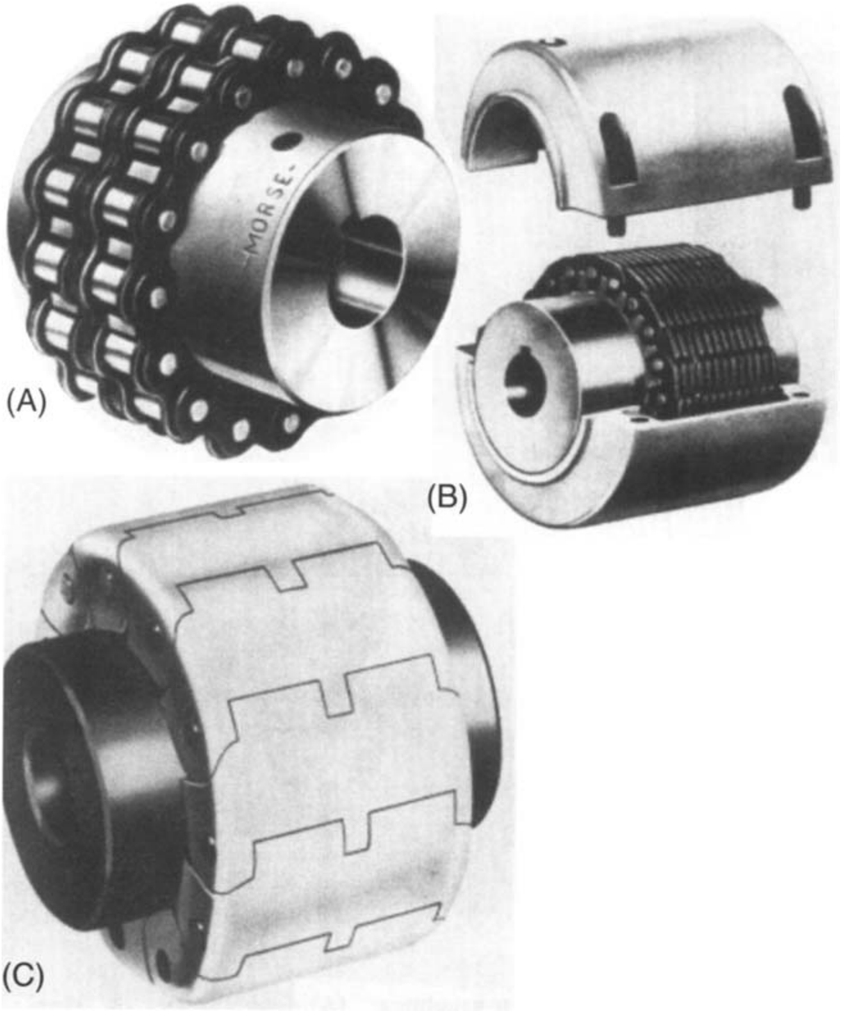


Figure 6.94 Chain couplings. (A) Double roller chain coupling. (B) Silent chain coupling. (C) Plastic chain coupling. (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

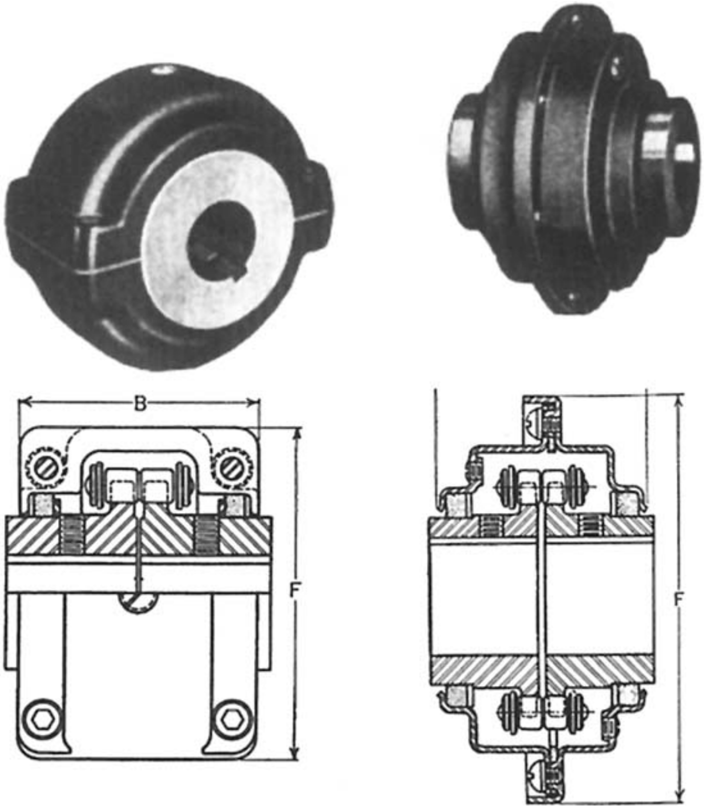
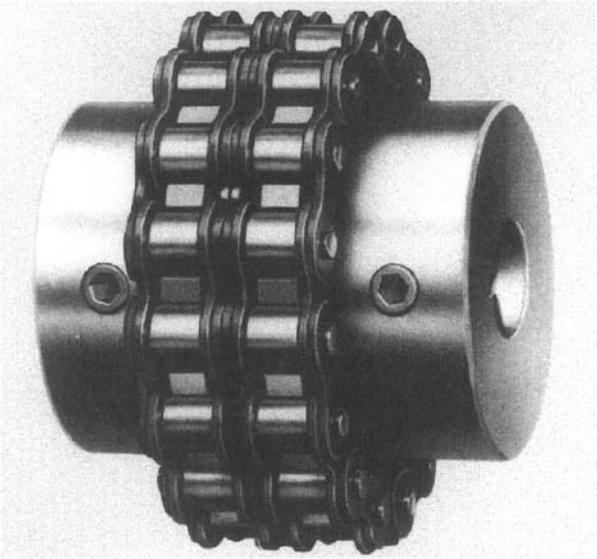


FIGURE 6.95 Typical chain coupling covers. (Courtesy of Link-Belt.)



Horsepower Ratings for Finished Bore and Minimum Bore Chain Couplings

Coupling Size	Below 50 RPM Max. Torque	Horsepower at Indicated Speeds											M R w Co
		50	100	200	300	400	600	900	*200	1500	1800	3600	
C4012	113	1.08	2.15	3.43	4.52	5.57	7.56	10.45	13.10	15.70	18.20	33.10	5,
C4016	200	1.90	3.81	6.07	8.00	9.86	13.40	18.50	23.20	27.80	32.20	58.50	5,
C4020	308	2.93	5.86	9.26	12.32	15.01	20.41	28.50	35.18	42.22	49.61	88.67	4,
C5016	384	3.66	7.32	11.70	15.35	18.90	25.70	35.53	44.50	53.30	61.90	112.00	4.
C5018	525	5.00	10.00	15.00	21.00	25.00	33.00	48.58	57.00	67.00	79.00	145.00	4.
C6018	910	8.70	17.30	27.60	36.38	44.90	60.90	84.20	105.00	126.00	147.00		3.
C6020	1050	10.00	23.00	36.00	41.98	58.00	79.00	97.16	135.00	165.00	192.00		3.
C8018	2027	19.30	38.60	61.40	81.05	99.80	135.00	187.57	234.00	281.00	326.00		2.
C8020	2625	25.00	50.00	80.00	104.96	130.00	175.00	242.90	302.00	365.00	430.00		2.
C10018	3644	34.70	69.40	111.00	145.70	180.00	244.00	337.30	422.00	506.00	587.00		1.
C10020	4495	42.80	85.60	136.00	179.73	221.00	300.00	415.95	517.00	621.00	708.00		1.

For maximum service life, a cover with proper lubrication is required for couplings selected with ratings to the right of the heavy line.Shaft and key stresses have not been considered in the above ratings.
***WARNING - Couplings with covers must not be operated beyond this speed.**

FIGURE 6.96 Typical sizes and horsepower ratings for chain couplings. (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

Table No. 12 **Load Classifications**

CLASS E Even Load	CLASS U Uneven Load	CLASS H Heavy Shock Load
Agitators for liquids	Beaters	Boat propellers
Blowers, centrifugal	Compressors, centrifugal	Compressors, reciprocating
Conveyor, belt or chain smoothly loaded	Conveyors, pulsating load	Crushers
Cranes	Elevators, pulsating load	Feeders, reciprocating
Elevator, smoothly loaded	Grinders, pulp	Machines, reversing or impact loads
Fans, centrifugal	Hoists	Mills, hammer
Generators	Kilns and Dryers	Oil Well Pumping Units
Line Shafts, even load	Line Shafts, uneven load	Presses
Machines, uniform load, non-reversing	Machines, pulsating load, non-reversing	Pumps, simplex or duplex, reciprocating
Pumps, centrifugal	Mills, ball, blooming, pebble, tube	Refuse Hogs
Screens, uniformly fed	Pumps, reciprocating	
Worm Gear Speed Reducers		

Table No. 13 **Service Factors**

Class	Characteristics of Driven Unit	Source of power		
		Electric Motor or Steam Turbine	Steam Engine or Gasoline Engine 4 or more Cyl.	Diesel or Gas Engine
E	Even load–8 hour/day service* Non reversing–low torque starting.	1	1 1/2	2
U	Uneven load–8 hour/day service* Moderate shock or torsional loads–Non reversing–This is the most common type of service.	1 1/2	2	2 1/2
H	Heavy shock load–8 hour/day service* High peak torsional loads–Reversing under load–Full load starting	2	2 1/2	3

* For 16 or 24 hour/day service use service factor for next higher class loading.

Note—For even load, stand by, seasonal or infrequent service the normal service rating of the coupling will determine its proper selection.

COUPLING SELECTION INFORMATION:

Application information: A finished bore chain coupling is required to drive a pulp grinder (1¾" shaft) from a 1800 RPM, 20 HP electric motor (1⅝" shaft) approximately 16 hours per day.

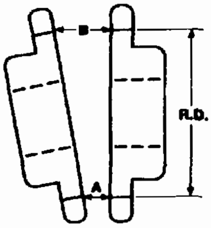
1. **Determine load classification and service factor.** Note that a pulp grinder is considered a Class "U" load but since it is to operate 16 hours per day, it must be classed as an "H" load. Table No. 13 indicates that the service factor for a Class H load driven by an electric motor is 2.

2. **Calculate the required horsepower.** Input Horsepower (20) x Service Factor (2) = Required Horsepower (40).

3. ***Determine the Coupling Size.*** *From Table No. 9, Page 6,* select the smallest coupling for 1800 RPM that meets the horsepower requirements. Verify bore requirements (1⅝" driver, 1¾" driven) for selected hub size. Select C5018 based on shaft requirements.

MISALIGNMENT

For maximum life, angular misalignment should not exceed ½". Refer to sketch to insure that .009 inches per inch of root diameter is not exceeded. This is equivalent to ½" of angular misalignment. Offset or parallel misalignment not to exceed 2% of chain pitch is recommended.



B-A = .009" X R.D.

FIGURE 6.97 Typical selection procedure for chain couplings. (Table numbers refer to the original Morse manual.) (Courtesy of Morse Industrial, Emerson Power Transmission Corporation.)

V. UNIVERSAL JOINTS*

A. Introduction

A universal joint is a special type of flexible coupling. There are three basic types:

1. The plain cross and bushing type
2. The constant-velocity type
3. The heavy-duty industrial type

Basically, they are all a form of the Hooke joint. The universal joint is the oldest known device used to connect to pieces of rotating equipment. Today, even with the use of sophisticated design procedures such as FEA (see Figure 6.98), use of high-strength materials, and use of the latest manufacturing equipment and processes, the universal joint is still one of the most power-dense connecting devices in the industry (Figure 6.99). These joints can handle up to 25,000,000 in.-lb of torque.

The universal joint can be used as a single joint or it can be used in pairs. The single universal joint cannot transmit power from one shaft to another shaft that is parallel to it, yet offset; the velocity of the second shaft varies harmonically with each revolution of the first. To analyze and use a universal joint, one must understand its kinematic relationships.

B. Applications of Universal Joints

Universal joints are used on automobiles, agricultural and locomotive vehicles, horizontal and vertical pumps, winches, paper processing equipment, conveyors, drilling rigs, and crane drives (see Figures 6.100 to 6.104).

C. Types of Universal Joints

1. Plain Cross and Bushing Type Joints

A very simple form is shown in Figure 6.105. It consists of two shaft shapes drilled at right angles and then connected through a third piece, a cross. Many variations of this joint are in use, some highly complicated. The plain cross and bushing is used on low-torque manual controls such as steering columns and on rotating applications that operate at less than 1000 rpm (see Figure 6.105). The bearing and cross design universal joint (see Figure 6.106) is used in applications that operate at up to 6000 rpm.

---

*Figures for this section are grouped together starting on p. 371.

2. Constant-Velocity Joints

Where uniform rotary motion is essential—even at high or unequal angles—a constant-velocity joint (see Figure 6.107) should be used. In this type of joint, power is transmitted through hardened steel balls that track in machined grooves of the outer and inner race of each assembly, while ball position is maintained by the cage member. This type of joint eliminates torsional or internal excitations inherent in other couplings. A common application for this type of joint is on off-road vehicles such as front-end loaders (see Figure 6.108A, railway maintenance equipment (see Figure 6.108B), earth-moving equipment, and agricultural equipment.

3. Heavy-Duty Universal Joints

There are four basic types of universal joints: (see Figure 6.109)

1. The block design universal joint (see Figure 6.110)
2. The split eye design universal joint (see Figure 6.111)
3. The one piece eye design (see Figure 6.112)
4. The split yoke design (see Figure 6.113)

There are many variations in the design of universal joints. Some of these are:

Welded fixed-shaft design, yoke type and block type (Figure 6.114): This joint is used when misalignment angles are fixed or where one end can slide in the bore.

Welded telescoping shaft, yoke type and block type (Figure 6.115): This joint is used when the misalignment angle changes in operation.

Flanged fixed shaft (Figure 6.116): This design allows for the replacement of universal joint heads without replacing the shaft.

Flanged universal joint (Figure 6.117): This joint allows for connection of very short shaft-to-shaft distances.

D. Design and Construction of Universal Joints

Of the two types of universal joints, the pin and block design capacity is usually based on the cross. This type of universal joint is constructed from a wide variety of materials. Usually forks, blocks, and pins are either through- or surface-hardened.

For universal joints that use bearings, there are many different types of construction: needle bearings, roller bearings, split eye yokes, and one-piece yokes. Usually, the bearings are through-hardened. The races and caps on which the bearings are mounted are usually case-hardened and ground.

Crosses and yokes are usually made of forged alloy steel. The yokes and/or the crosses on large industrial universal joints are sometimes made of cast steel.

Several different kinds of flange connections are used with universal joints (see figure 6.118).

E. Kinematic Relationships of Universal Joints

1. Single Universal Joint

Two shafts inclined at angles to one another are connected by a single universal joint, and the driving shaft is driven at the uniform angular velocity ω_1 (Figure 6.119). With this single universal joint system, the driven shaft rotates in a nonuniform manner with variable angular velocity.

The amplitude of the sinusoidal variation of angular velocity, or of the angles of lead and lag of the driven shaft, are largely dependent on the angle of deviation. This characteristic of universal joints, termed Cardan error, must be allowed for in the dimensioning and application of universal joints. It is clear from Figure 6.120 that with one revolution of shaft section 1, four variations of angular velocity ω_2 occur. This means that shaft section 2 passes through the acceleration and deceleration maxima twice in every revolution. With a large amount of angle of deviation and at high rotational speeds, considerable inertial force can become effective. It is thus apparent that a single universal joint can be employed only in applications where low forces are to be transmitted at low speeds and with small angles of deviation and where the nonuniformity of rotary motion in the output shaft is of subordinate significance.

Symbols and their meanings:

G_1 = joint between shaft section 1 and shaft section 2
G_2 = joint between shaft section 2 and shaft section 3
β_1 = angle between shaft section 1 and shaft section 2 on the plane formed by the two shaft sections (deg)
β_2 = angle between shaft section 2 and shaft section 3 on the plane formed by two shaft sections (deg)
β_S = angle projected onto the vertical plane (deg)
β_W = angle projected onto the horizontal plane (deg)
β = angle of deviation (deg)
α_1 = angle of rotation of shaft section 1 (deg)
α_2 = angle of rotation of shaft section 2 (deg)
α_3 = angle of rotation of shaft section 3 (deg)
ϕ = differential angle (lead and lag angle) (deg)
ω_1 = angular velocity of shaft section 1
ω_2 = angular velocity of shaft section 2
ω_3 = angular velocity of shaft section 3
U = coefficient of cyclic variation

Assuming that the yoke position of joint 1, shown in Figure 6.120 as $\alpha_1 = 0$, is the zero position of angle of rotation α_1, the following relationships apply:

$$\phi = \alpha_1 + \alpha_2$$

$$\frac{\tan \alpha_1}{\tan \alpha_2} = \cos \beta$$

$$\tan \phi = \frac{\tan \alpha_1 (\cos \beta - 1)}{1 + \cos \beta \tan^2 \alpha_2}$$

This yields for the angular velocities and torque values of shaft sections 1 and 2:

$$\frac{\omega_2}{\omega_1} = \frac{\cos \beta}{1 - \sin^2 \beta \sin^2 \alpha_1}$$

$$\frac{M_2}{M_1} = \frac{\omega_2}{\omega_1}$$

$$\text{Maximum } \frac{\omega_2}{\omega_1} = \frac{1}{\cos\beta} \qquad \text{at } \alpha_1 = 90^\circ \text{ and } 270^\circ$$

$$\text{Maximum } \frac{\omega_2}{\omega_1} = \cos\beta \qquad \text{at } \alpha_1 = 90^\circ \text{ and } 180^\circ$$

$$\text{Maximum } \frac{M_2}{M_1} = \frac{1}{\cos\beta} \qquad \text{at } \alpha_1 = 90^\circ \text{ and } 270^\circ$$

$$\text{Maximum } \frac{M_2}{M_1} = \cos\beta \qquad \text{at } \alpha_1 = 0^\circ \text{ and } 180^\circ$$

For a comparison of nonuniformity, the coefficient of cyclic variation U has been introduced (see Figure 6.121):

$$U = \frac{\omega_{2\,\max} - \omega_{2\,\min}}{\omega_1} = \tan\beta \sin\beta$$

$$\tan\phi\,(\max) = \pm \frac{1 - \cos\beta}{2\sqrt{\cos\beta}}$$

2. Double Universal Joint

The statements made in Section 5.A.1 show that for a single universal joint with a given deviation angle, shaft section 2 always rotates at an irregular angular velocity. If, however, two universal joints are correctly arranged in a "Z" or "W" arrangement as shown in Figure 6.122, a perfect compensation of nonuniform motion of shaft sections 2 and 3—the output shaft—synchronizes with shaft section 1—the input shaft. Synchronous running of shaft sections 1 and 3 is obtained where:

1. All shaft sections of the universal joint lie on one plane (Figure 6.123A).
2. Yokes 1 and 2 of section 2 lie on one plane (Figure 6.123B).
3. The deviation angles β_1 and β_2 of universal joints 1 and 2 are equal (Figure 6.123C).

These conditions ensure that joint 2 works offset in phase by 90° to joint 1 and completely cancels out the influence of universal joint 1. With such an ideal universal joint shaft arrangement, perfect kinematic balance is possible. In practice, we always seek this perfect balance. If any one of the three conditions is not fulfilled, the universal shaft will not work homokinetically. In such cases, it is advisable to consult the universal joint manufacturer, since certain influences can be compensated for by specific measures.

3. Determination of Deviation Angle β

If a universal joint lies spatially on a plane Z as illustrated in Figure 6.124, angles β_W and β_S can be determined from the top and side views of the drawing. For the resultant deviation angle β on plane Z, which is decisive for the dimensioning of the universal joint, the formula

$$\tan \beta = \sqrt{\tan^2 \beta_W + \tan^2 \beta_S}$$

applies. Its graphic representation is shown in Figure 6.125.

F. Rating and Selection of Universal Joints

There are two types of universal joint ratings:

1. *Strength rating* is based on the endurance strength or yield strength of the weak link of the universal joint.
2. *Life rating* is based on the B-10 life for the bearing and/or the life rating of the sliding shaft if one is used. A B-10 life is defined as the minimum life expectancy for a 90% probability of survival. The average bearing life would be approximately five times the B-10 life.

Usually, the weakest link of a universal joint is the cross. The strength of the cross is a function of the bending strength of the trunions of the cross.

G. How Universal Joints Affect the System

1. Axial Forces for Loads on Connected Equipment

$$F = \frac{T\mu}{R \cos \beta}$$

where

T = operating torque (in.-lb)
μ = coefficient of friction, 0.15

β = misalignment angle
R = pitch radius of length compensating spline (in.)

2. Radial Forces for Loads on Connected Equipment

A misaligned universal joint produces cyclic forces on connected shafts and bearing. The following equations calculate the maximum amplitude of those forces (see Figure 6.126).

α = angle of rotation (deg)
T = torque (in.-lb)
β_1 = angular misalignment of end A
β_2 = angular misalignment of end B
L = length between joint centers
A, B, C, D = radial bearing forces at 0°
$A_1, B_1, C_1, D_1, C_1{}^1, D_1{}^1$ = radial bearing forces at 90°
E, F = distance from joint center to the first bearing
G, H = distance between bearing of equipment at 0°

Note: Use addition (+) for W arrangement; use subtraction (–) for Z arrangement. At 90°, for W and Z arrangements:

$$A_1 = B_1 = \frac{T \times \tan \beta_1}{G}$$

$$C_1 = D_1 = \frac{T \times \sin \beta_2}{H \cos \beta_1}$$

H. Failure Mode for Universal Joints*

1. Types of Failures

Universal joints failures generally fall into two categories:

Structural failures, resulting from fatigue or overload
Failure from wear

There is a easy way to distinguish overload from fatigue. Simply think of overload failure as a sudden snapping and breaking of a matchstick (Figure 6.127), whereas fatigue can be compared to the flexing of a wire coat hanger (Figure 6.128). It takes repeated bending back and forth before the metal finally cracks and breaks.

---

*Parts of this section and many of the illustrations are from "Straight Talk About U-Joints," courtesy of Rockwell International.

2. Causes of Fatigue Failures

Fatigue failure occurs when driveshaft parts are operated under continuous long-term high-load conditions that exceed the endurance limit of the component. Exceeding the endurance limit means that the maximum operating load seen by the part is greater than the set endurance limit. This load condition must occur repetitively before the part finally fails. These loads are less than the yield limit of the part.

a. How to Detect Fatigue Failure of Driveshafts. Fatigue failures begin as tiny cracks that develop on the surface of a part over a period of time.

1. *Before total failure:* Cracks may start out almost invisible to the naked eye. Such a condition is difficult to spot without some type of nondestructive inspection (dye penetrate, magnetic particle inspection, etc.). Or one can wait until the part totally fails (Figure 6.129).
2. *After total failure: Beach marks* describe a failure that leaves a smooth metal surface with the appearance of contoured lines, much like the lines left on a sandy beach by wave action. When this condition is observed, it means the fatigue began as a small initial crack and then continued on to fatigue and then to final breakage (Figure 6.130). When *radial or circular lines* appear throughout the failure surface along with a "cone"-shaped fracture, this indicates that the failure was fatigue rather than overload (Figure 6.131).

b. Types of Overload Failures

1. *Shaft or tube overload:* Sometimes when an overload occurs, the driveshaft center tube will twist permanently (See Figure 6.132). Or tubes crack through their weld (see Figure 6.133). Or center shaft fails from torsion. This usually results in a rather smooth, flat surface failure (Figure 6.134).
2. *Cross damage:* Under high overload the trunions of the cross break away from the body of the cross (Figure 6.135).
3. *Spline shaft failure:* High load can cause failure through the

spline. The failure is usually at an angle across the spline (Figure 6.136).

4. *Yoke failure:* This type of failure is unusual but if the load is high enough it can occur (Figure 6.137).
5. *Block Failure:* Like yoke failures this type is unusual (Figure 6.138).

c. Types of Surface Wear

1. *Premature surface wear:* This is usually caused by one or more of the following conditions: lack of lubrication, incorrect lubrication, contaminated lubricants, inadequate or infrequent lube intervals, improper parts installation, and constant "high" operating loads.
2. *Normal Wear:* Even properly lubricated driveshafts will wear out in time. Normal wear of a universal can be compared to wear of automobile brakes. Properly maintained and properly applied brakes gradually wear out over a period of time. That is normal. Of course, riding the bakes or making repeated high-speed stops will wear out the linings much more rapidly.
3. *Brinelling:* This is a condition in which the roller wears grooves into the trunnion, cap, or cup (or liner if used). Replacement of the cross, cup, or cap (or the liners) is the solution (Figure 6.139).
4. *False brinelling:* As the name implies, this condition is *not* a true case of brinelling. Rollers sometimes polish the trunnion or needle cups so that they appear to have brinelling—even under low loads. You can determine false brinelling by rubbing your thumbnail across the surfaces. If you do not feel any roughness or indentations, then the condition you see is simply false brinelling. False brinelling does no damage to the surfaces and the part is still suited for continued service (Figure 6.140).
5. *Pitting:* Corrosion is the major cause of pits or craters that develop on the surface of the trunions and rollers. Excessive pitting causes an increase in friction that results in rapid surface wear (Figure 6.141).
6. *Spalling:* Up to now we have talked purely about surface

> wear. Spalling is another type of surface deterioration, related to metal fatigue. This term is used to describe metal that flakes off the surface of the trunnion. Spalling is caused by metal fatigue, not wear from friction. It develops after high loads have been imposed on the bearings, crosses, cups, caps, or liners millions of times. This breakdown of the structure takes place below the surface (subsurface) and causes the metal to "flake" or peel off (Figure 6.142).

Figure 6.143 lists some of the most common problems, their probable causes, and corrective measures for universal joints.

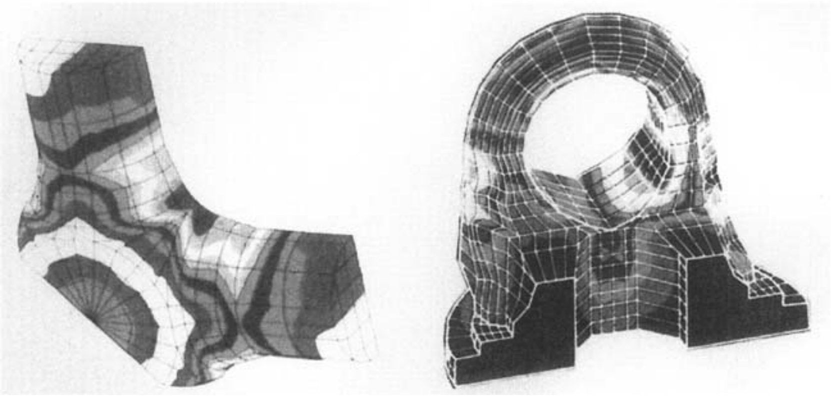


FIGURE 6.98 Yoke FEA. (Courtesy of Voith Transmit GmbH.)

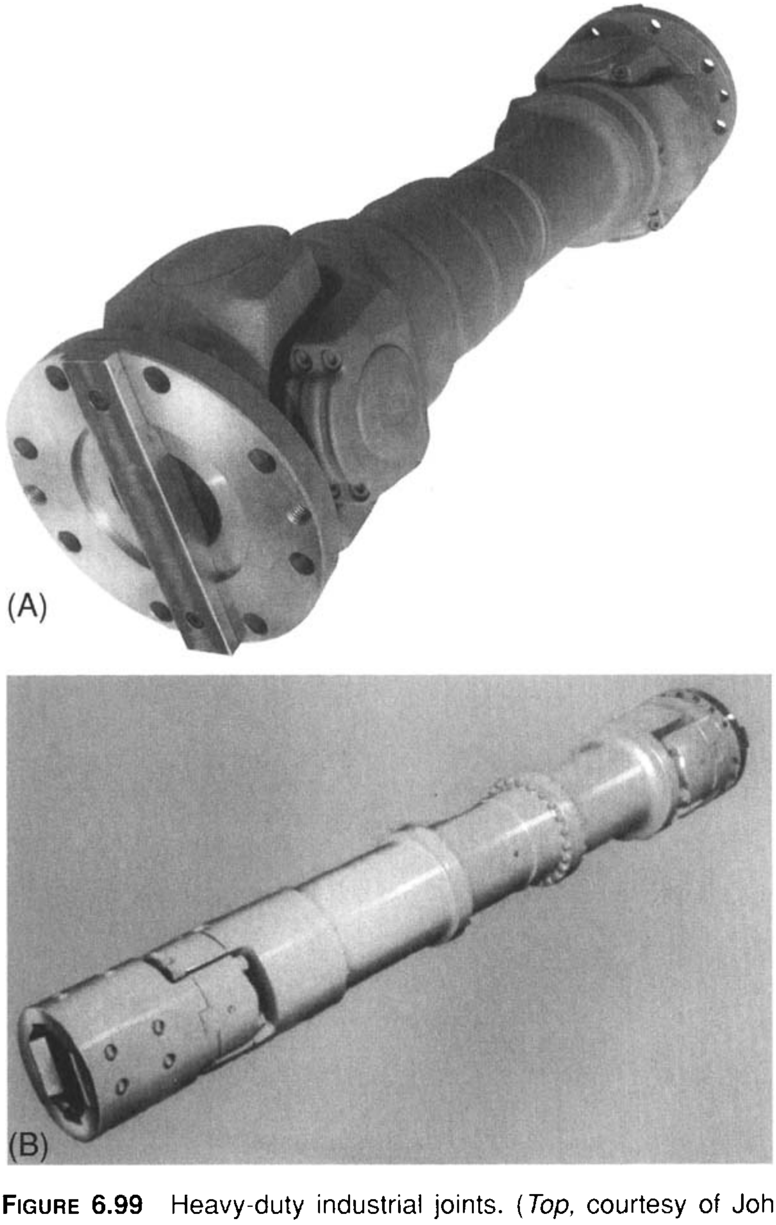


Figure 6.99 Heavy-duty industrial joints. (*Top*, courtesy of Johnson Power Ltd.; *bottom*, courtesy of Kop-Flex, Emerson Power Transmission Corporation.)

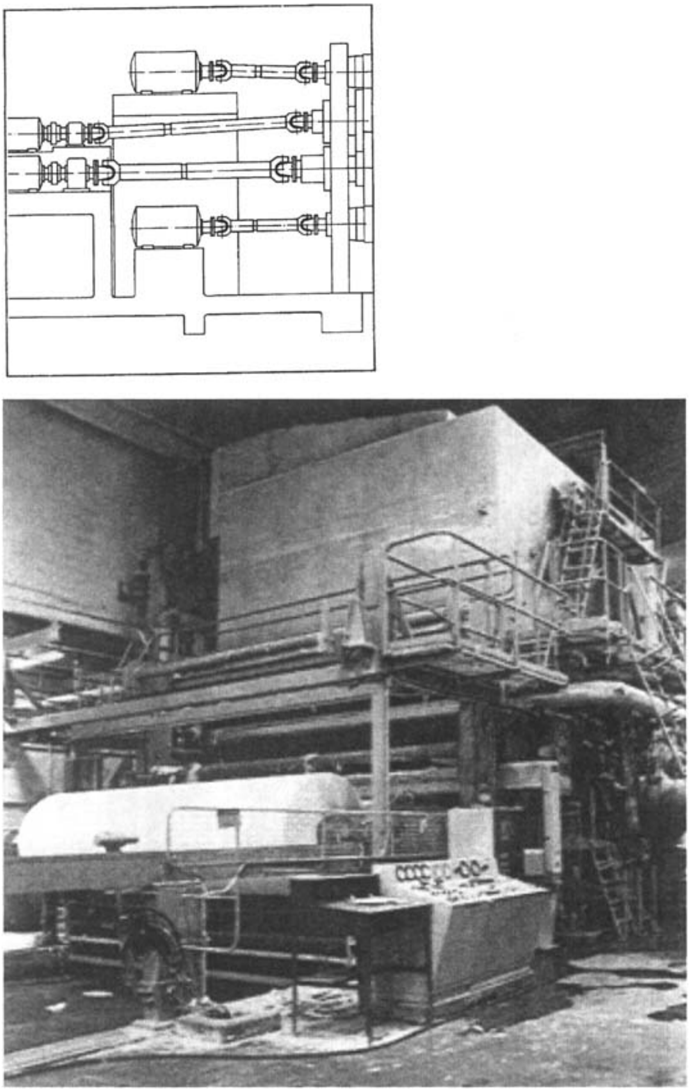


Figure 6.100 Paper machine application. (Courtesy of Voith Transmit GmbH.)

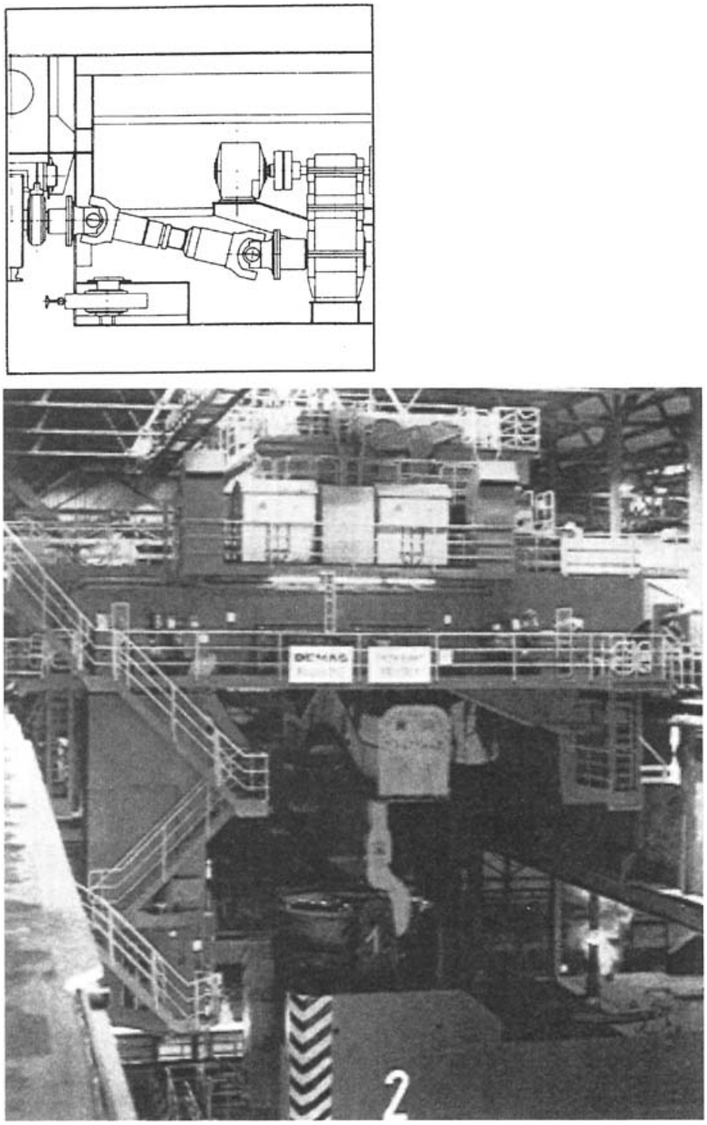


FIGURE 6.101 Crane application. (Courtesy of Voith Transmit GmbH.)

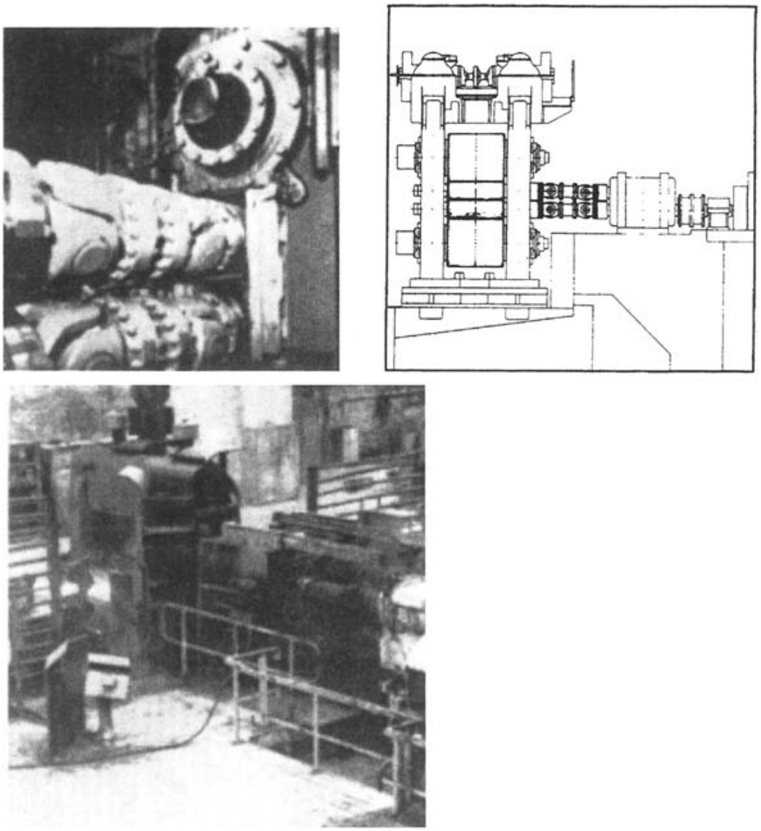


Figure 6.102 Cold-rolling mill application. (Courtesy of Voith Transmit GmbH.)

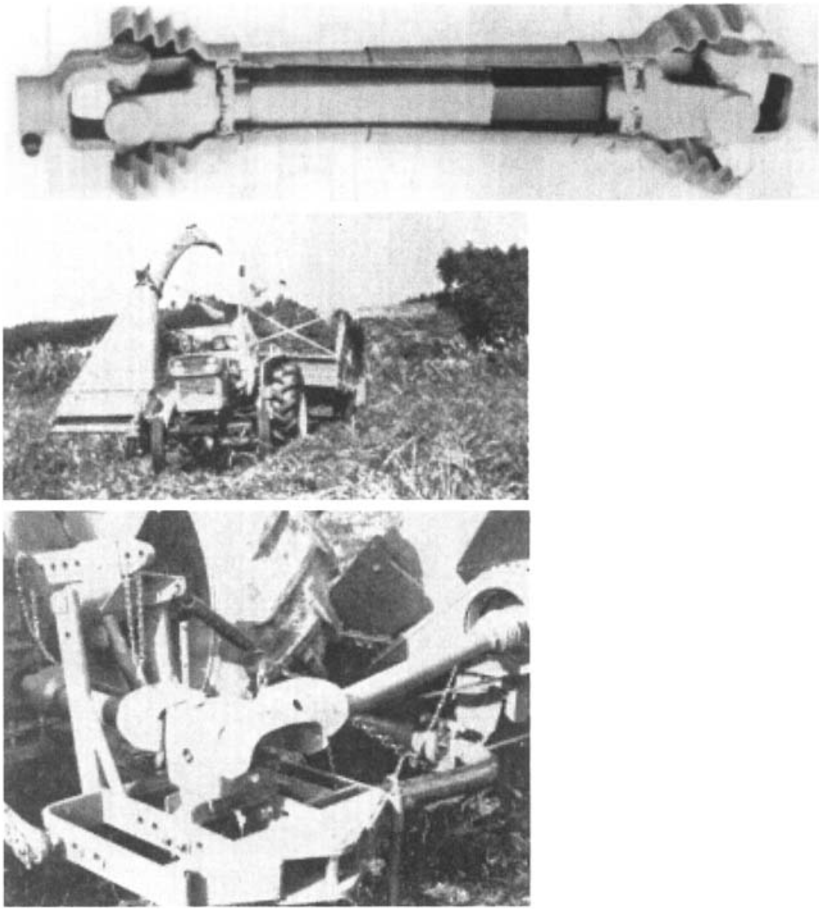


Figure 6.103 Power-takeoff application. (Courtesy of Koyo Seiko Co, Ltd.)

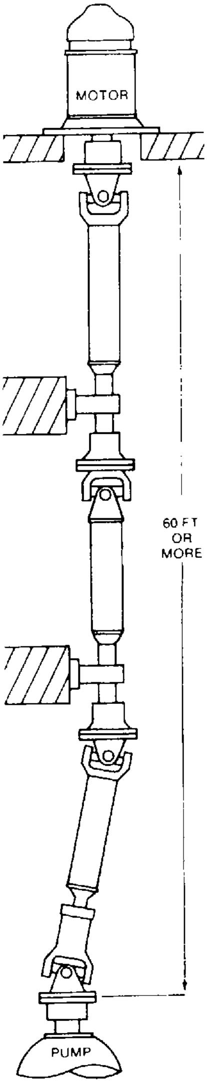


FIGURE 6.104 Vertical pump application.

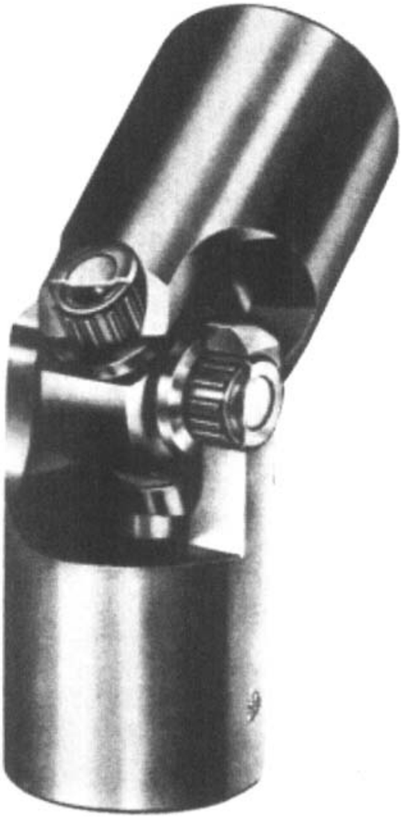


FIGURE 6.106 Bearing and cross design. (Courtesy of Fenner America Ltd.)

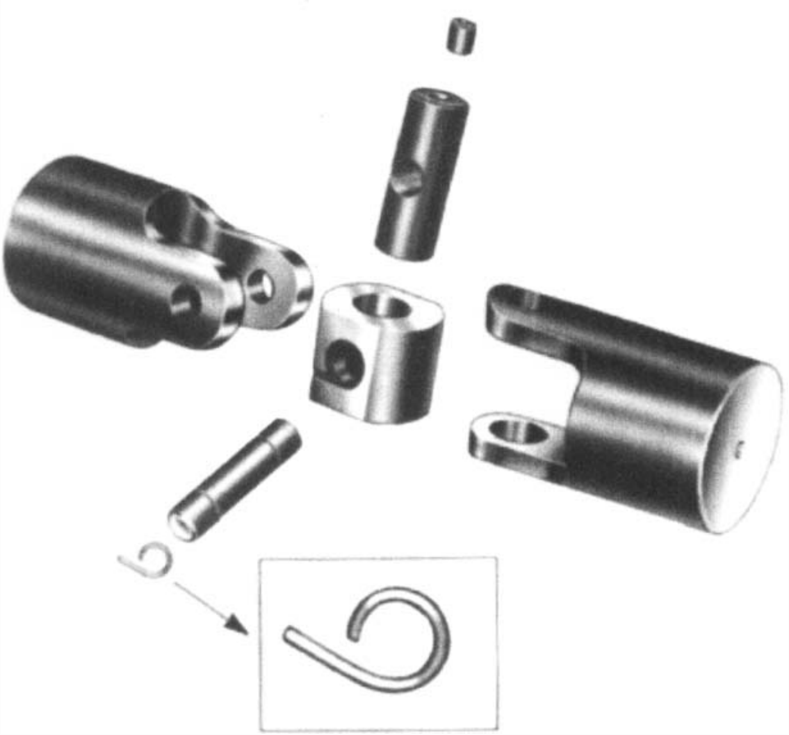


FIGURE 6.105 Plain cross and bushing joint.

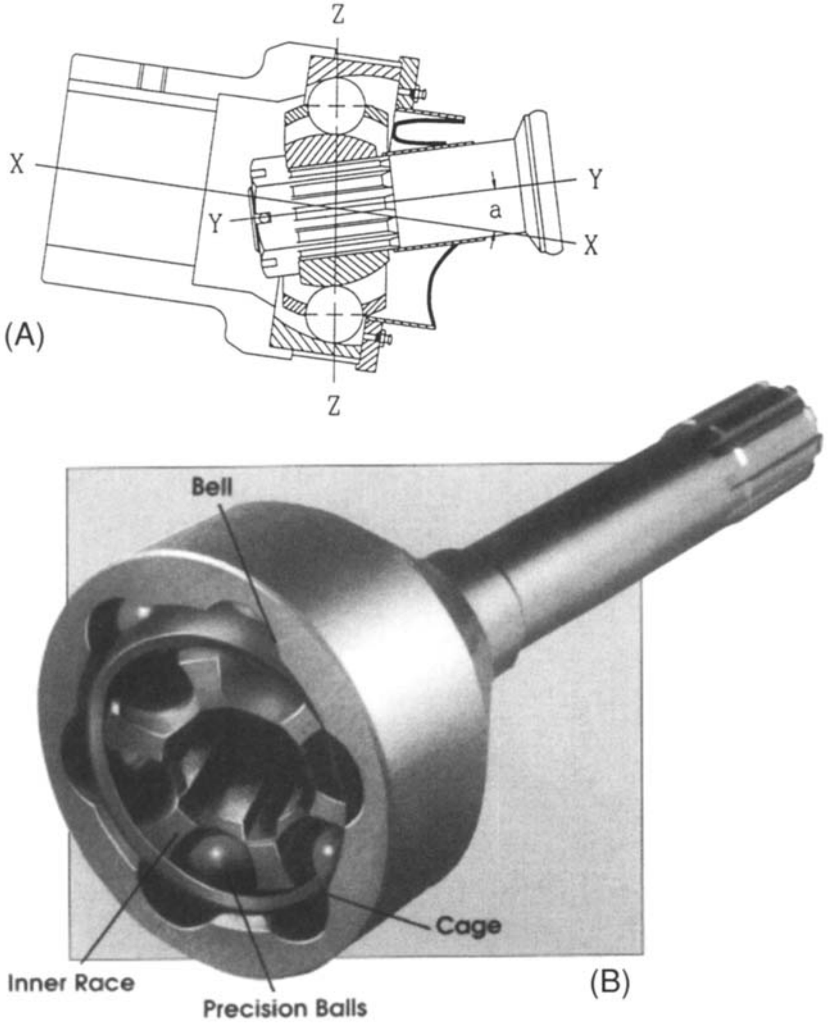


FIGURE 6.107 Constant-velocity joints. (Courtesy of Convel, Inc.)

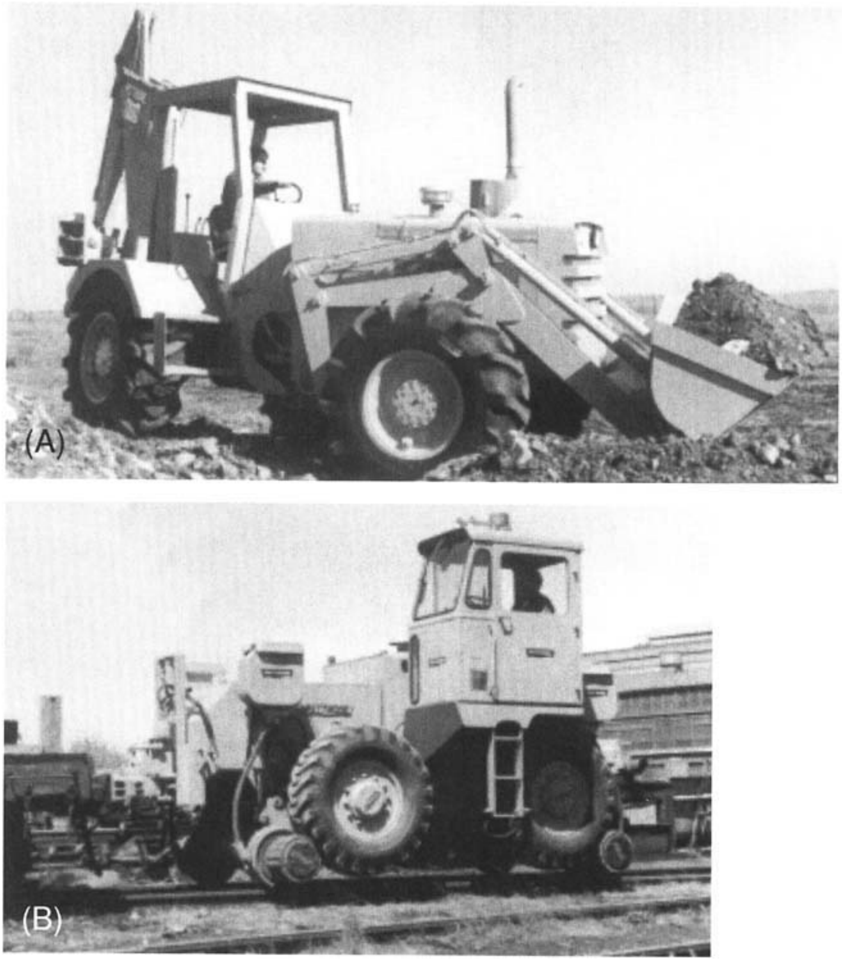
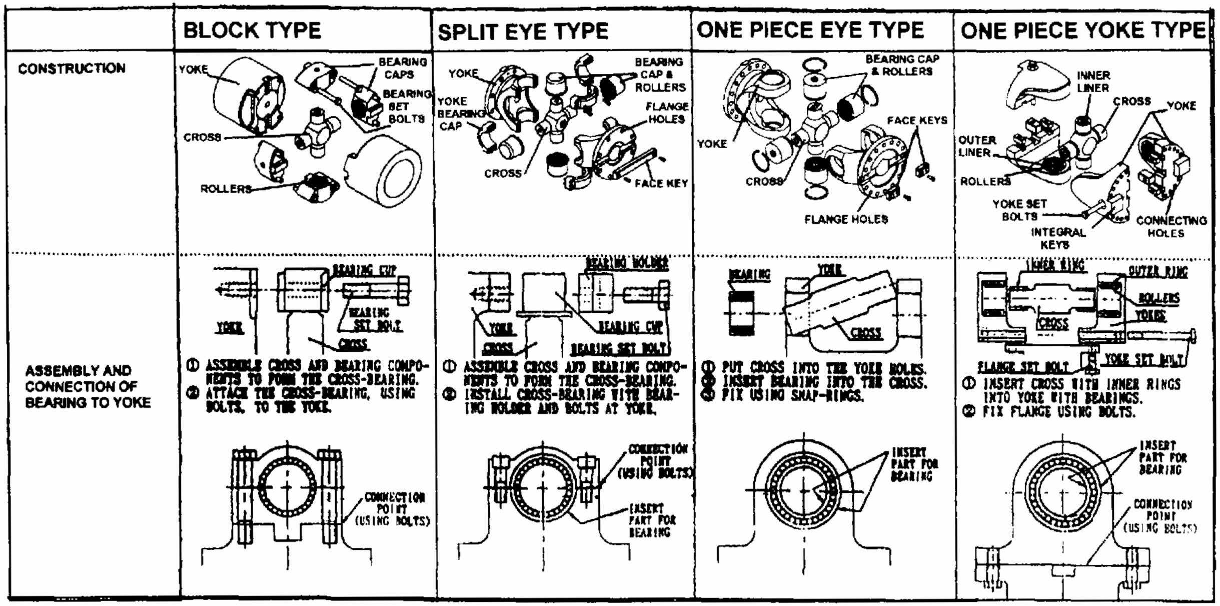


Figure 6.108 (A) Front end loader application. (B) Railway maintenance equipment application. (Courtesy of Convel, Inc.)



① ASSEMBLE CROSS AND BEARING COMPONENTS TO FORM THE CROSS-BEARING.
② ATTACH THE CROSS-BEARING, USING BOLTS, TO THE YOKE.

① ASSEMBLE CROSS AND BEARING COMPONENTS TO FORM THE CROSS-BEARING.
② INSTALL CROSS-BEARING WITH BEARING HOLDER AND BOLTS AT YOKE.

① PUT CROSS INTO THE YOKE HOLES.
② INSERT BEARING INTO THE CROSS.
③ FIX USING SNAP-RINGS.

① INSERT CROSS WITH INNER RINGS INTO YOKE WITH BEARINGS.
② FIX FLANGE USING BOLTS.

FIGURE 6.109 Types of heavy duty universal joints.

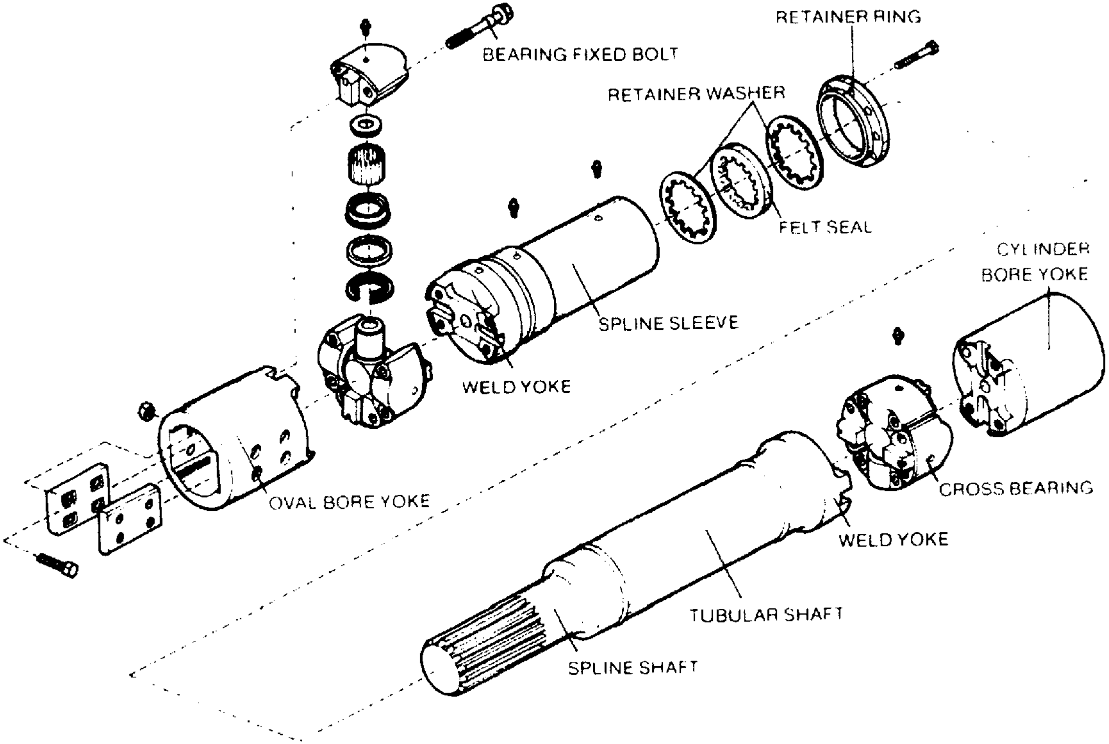


FIGURE 6.110 Block type universal joint. (Courtesy of Kop-Flex, Emerson Power Transmission Corporation.)

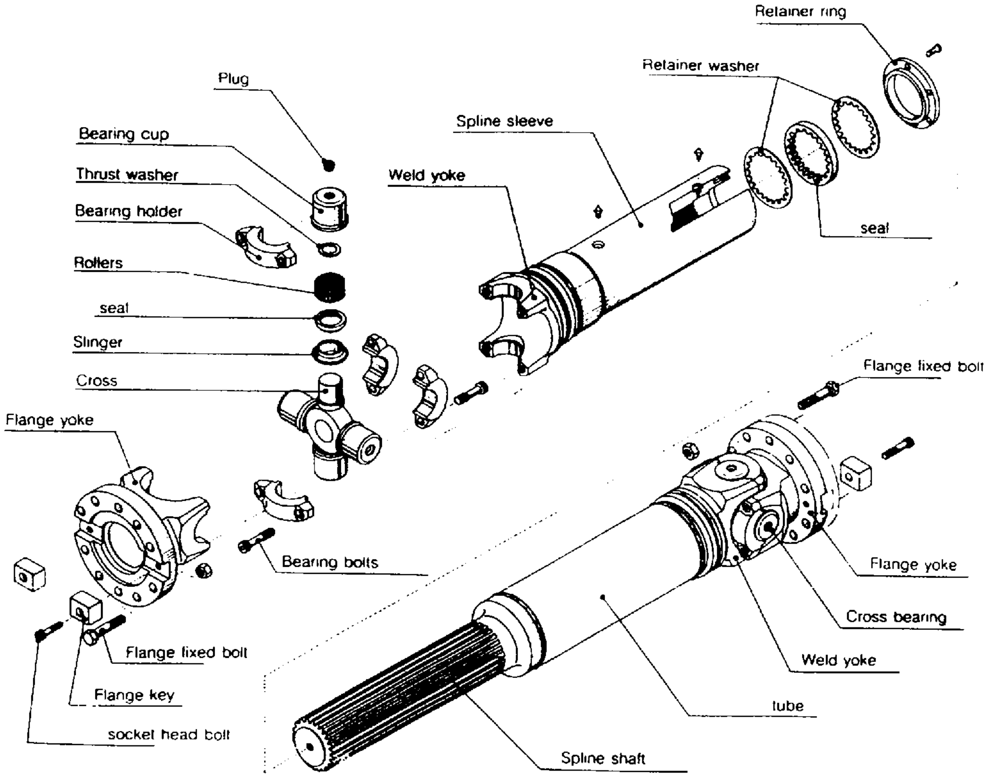


FIGURE 6.111 Split eye universal joint. (Courtesy of GWB.)

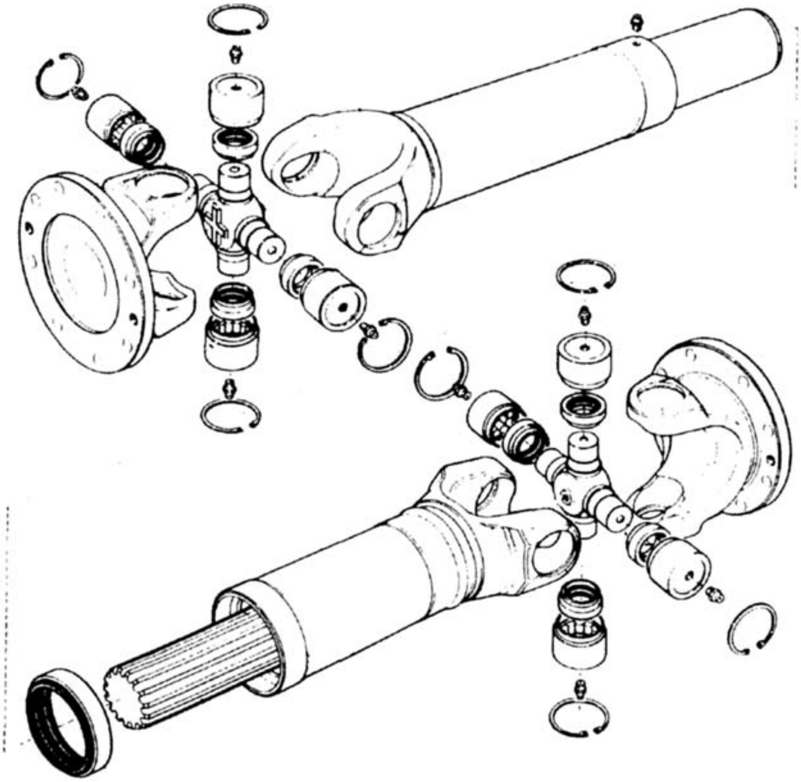


Figure 6.112 One-piece eye design. (Courtesy of Voith Transmit GmbH.)

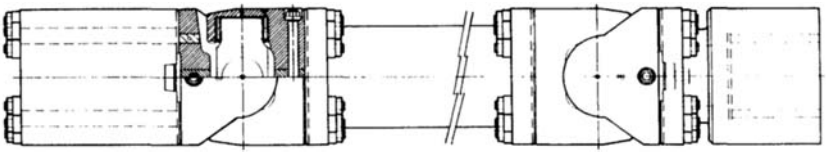


Figure 6.113 Split yoke design. (Courtesy of Morgan Construction.)

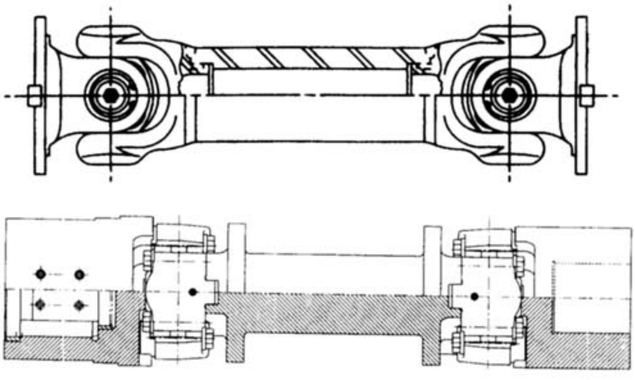


FIGURE 6.114 Fixed-length designs.

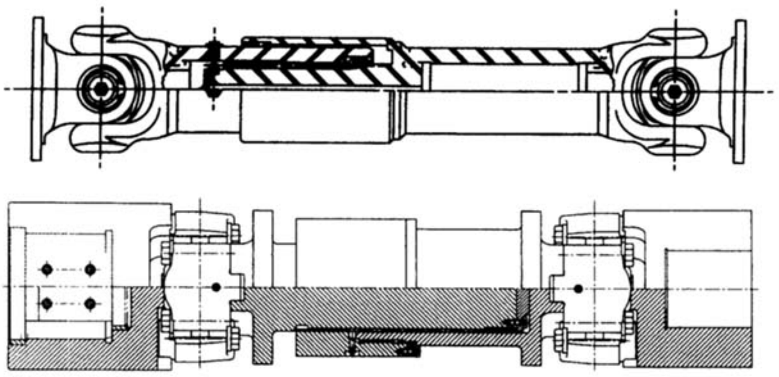


FIGURE 6.115 Telescoping design.

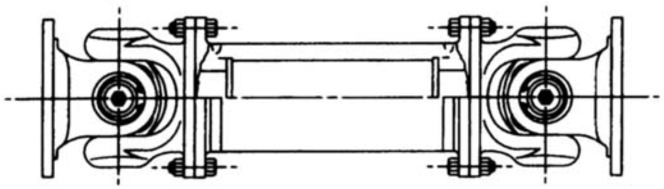


FIGURE 6.116 Flanged fixed-shaft design.

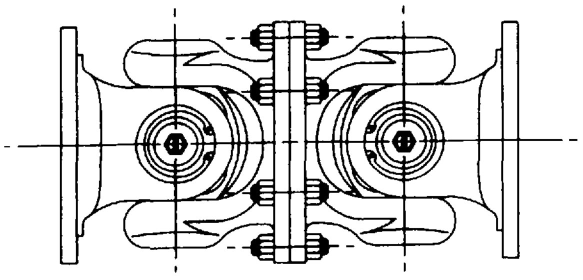


FIGURE 6.117 Close-coupled flanged design.

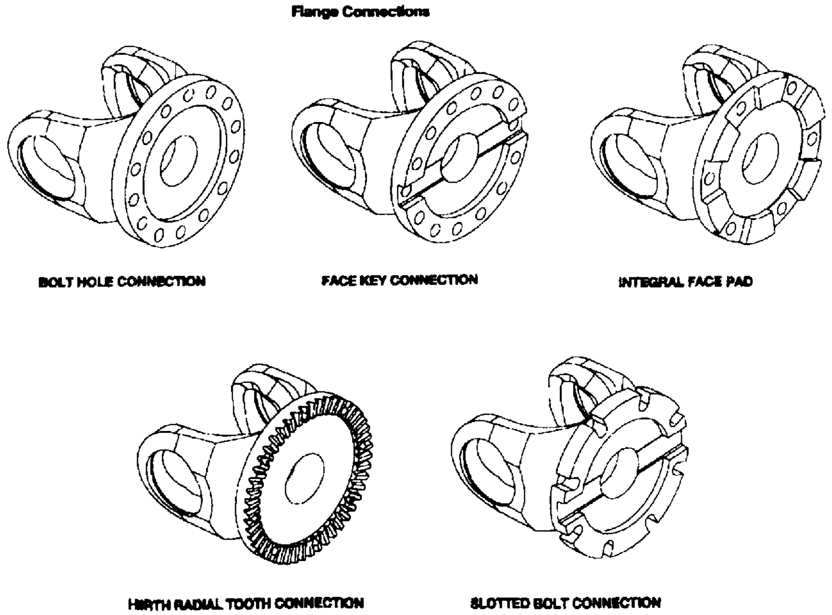


FIGURE 6.118 Variation of flange designs for universal joints. (Courtesy of Ameridrives International.)

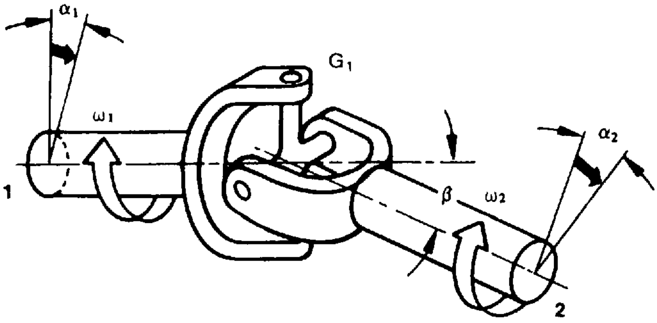


FIGURE 6.119 Simple universal joint.

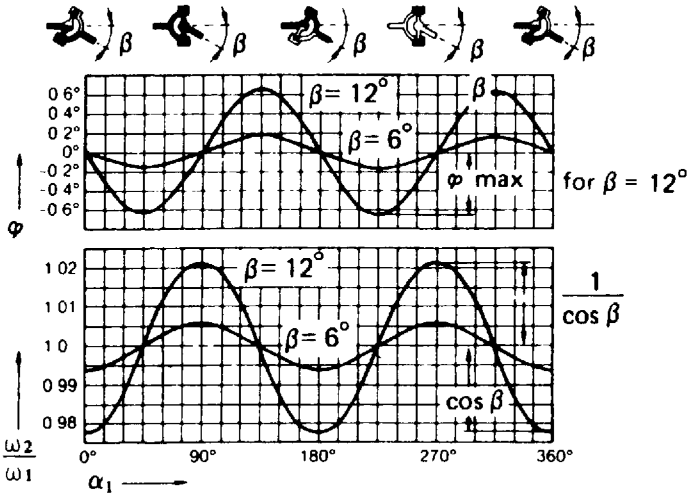


FIGURE 6.120 Cardan error.

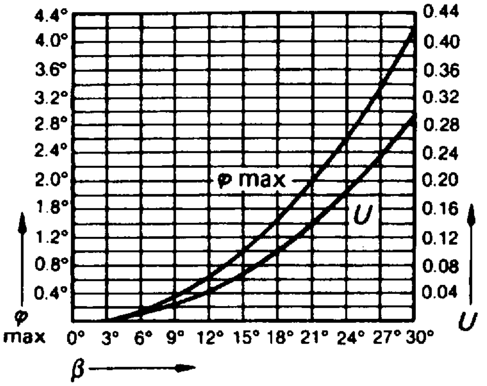


FIGURE 6.121 Coefficient of cyclic variation.

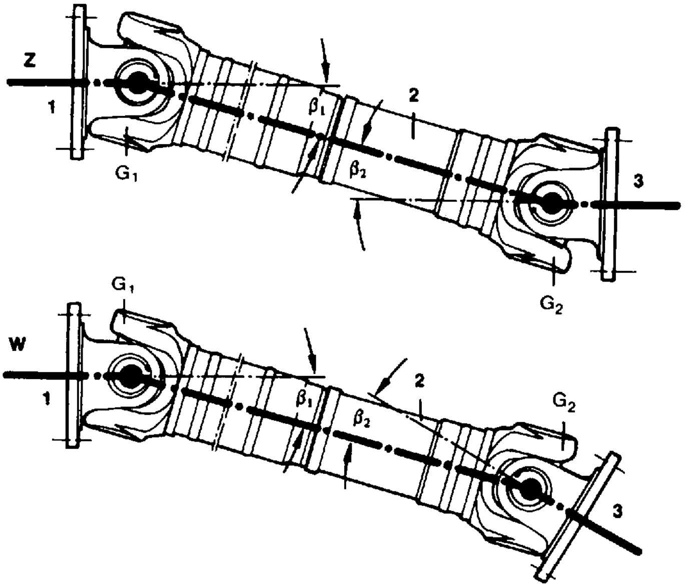


FIGURE 6.122 Z and W arrangements of a universal joint.

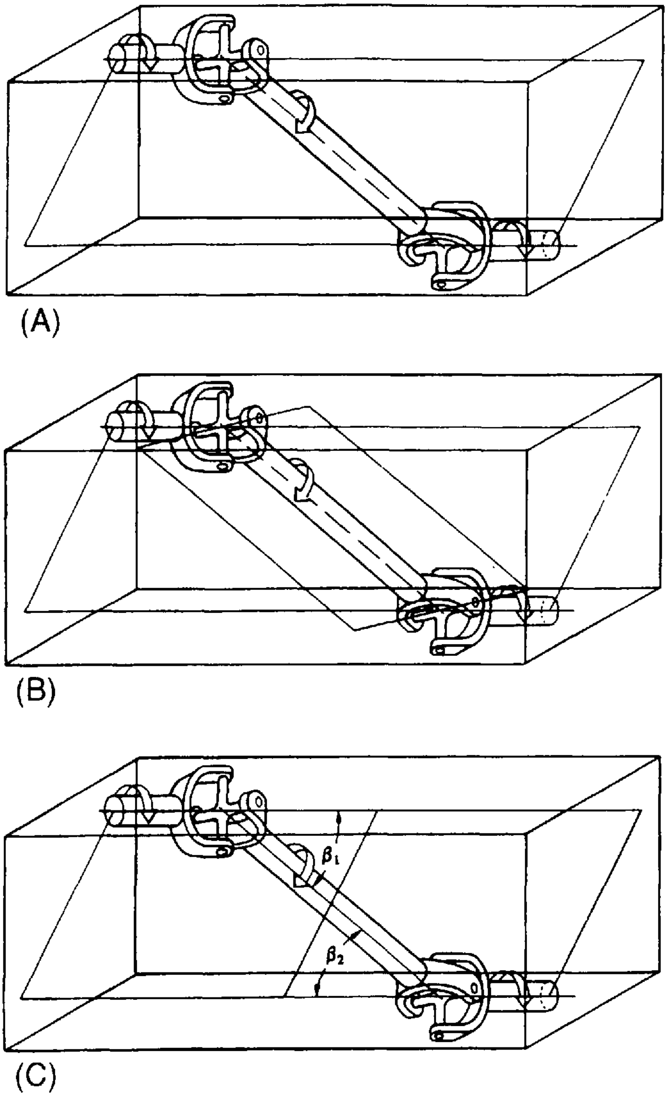


FIGURE 6.123 Various misalignment phases of a universal joint.

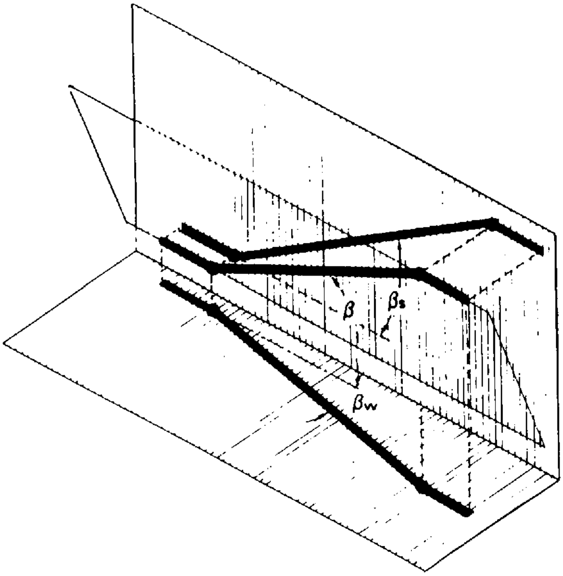


FIGURE 6.124 Universal joint spatial planes.

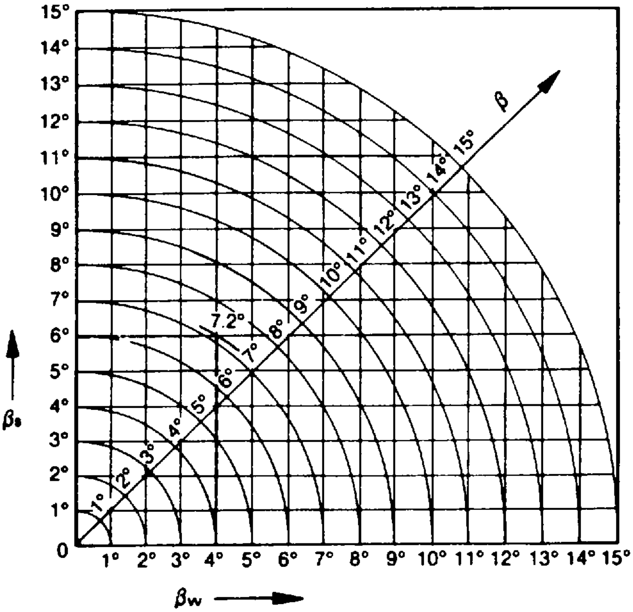


FIGURE 6.125 Combined misalignment for a universal joint.

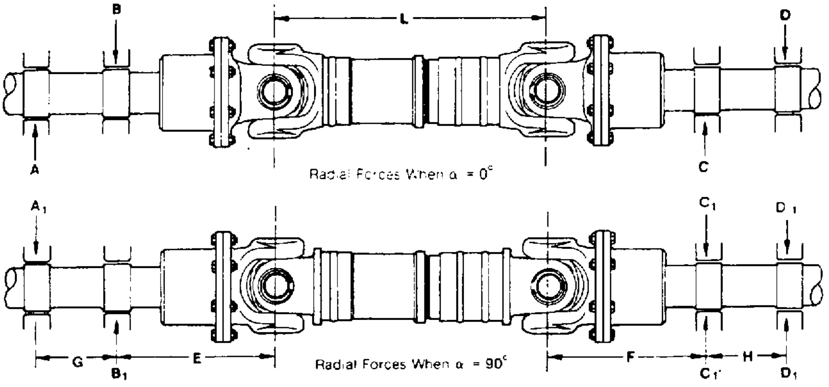


Figure 6.126 Universal joint reaction forces.

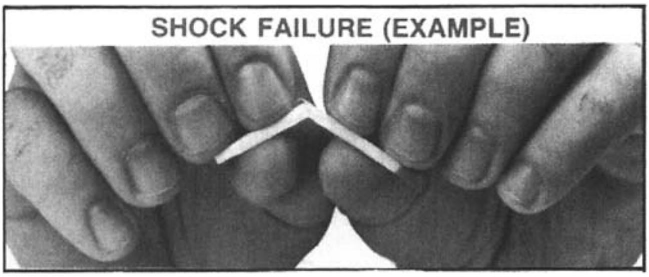


Figure 6.127 Shock failure (example).(Courtesy of Rockwell International.)

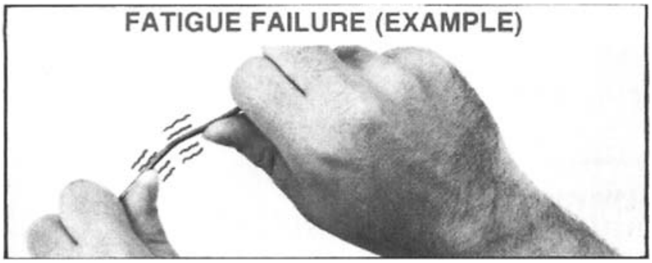


Figure 6.128 Fatigue failure (example). (Courtesy of Rockwell International.)

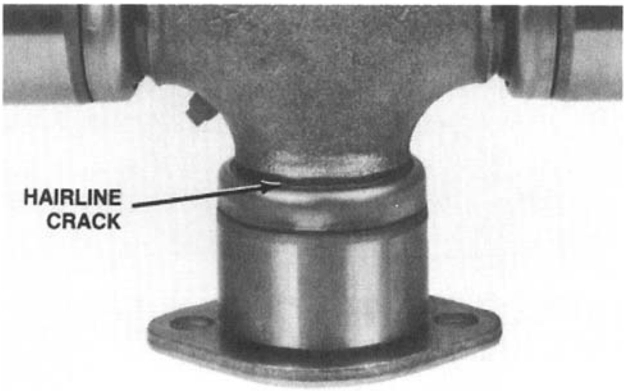


FIGURE 6.129 Hairline crack in cross. (Courtesy of Rockwell International.)

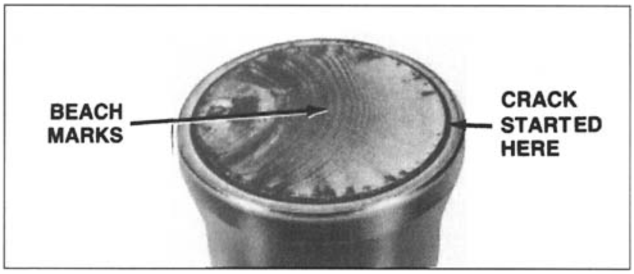


FIGURE 6.130 Beach marks—typical fatigue failure. (Courtesy of Rockwell International.)

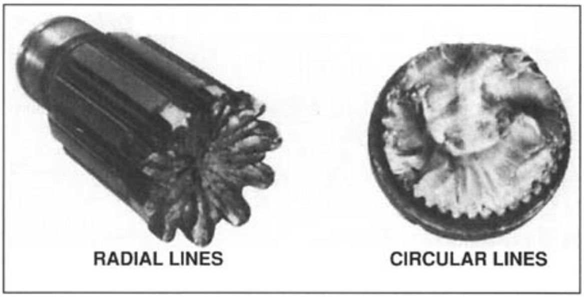


FIGURE 6.131 Typical fatigue failure of splined shaft. (Courtesy of Rockwell International.)

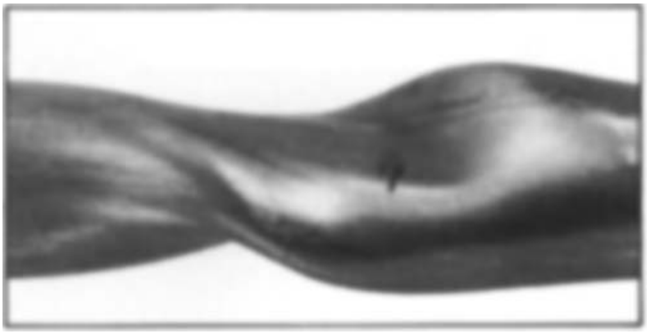


FIGURE 6.132 Shock failure of tubular shaft. (Courtesy of Rockwell International.)

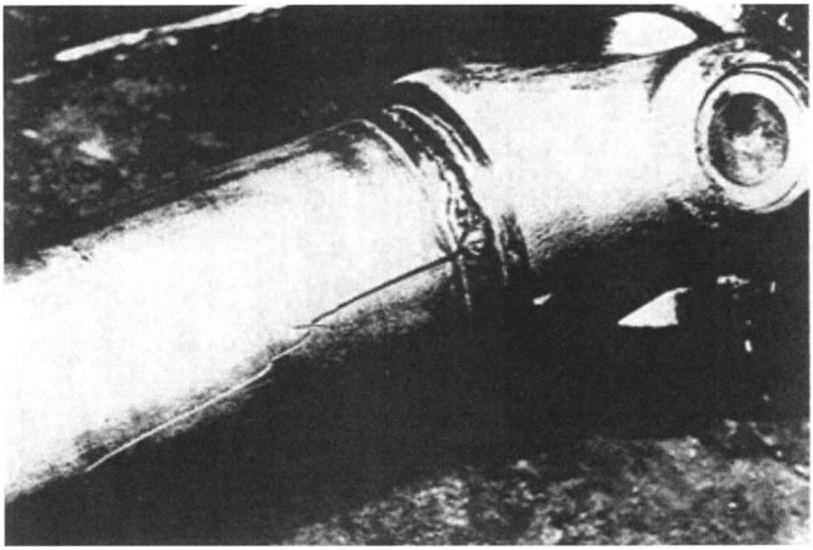


FIGURE 6.133 Failure of tube through welded area.

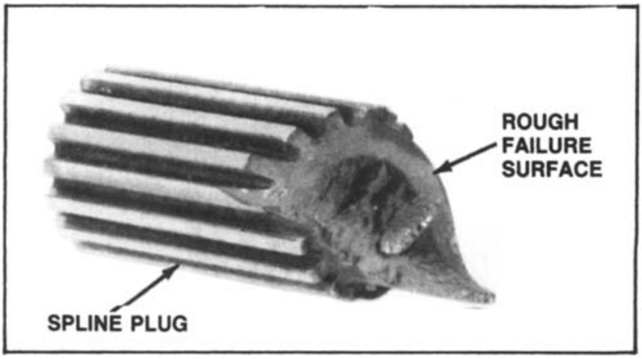


Figure 6.134 Shock failure of shaft. (Courtesy of Rockwell International.)

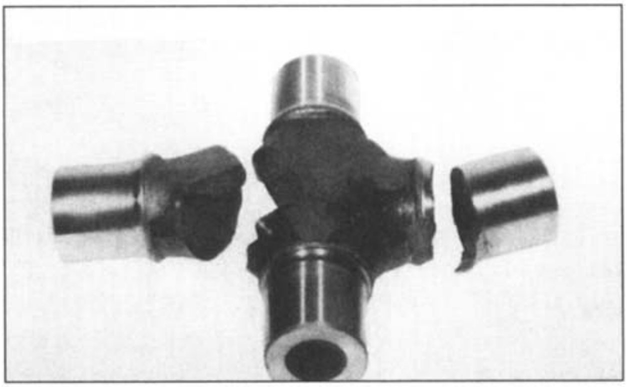


Figure 6.135 Cross failure from overload. (Courtesy of Rockwell International.)

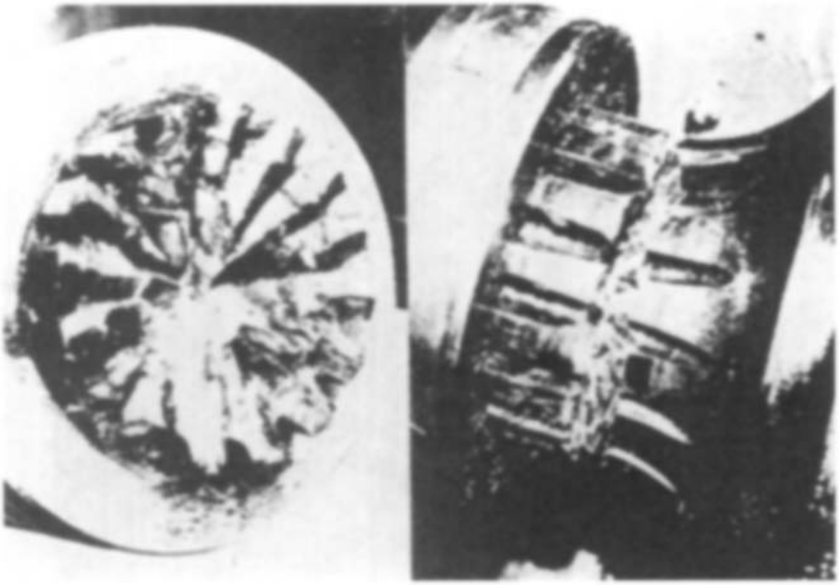


FIGURE 6.136 Spline shaft failure from overload.

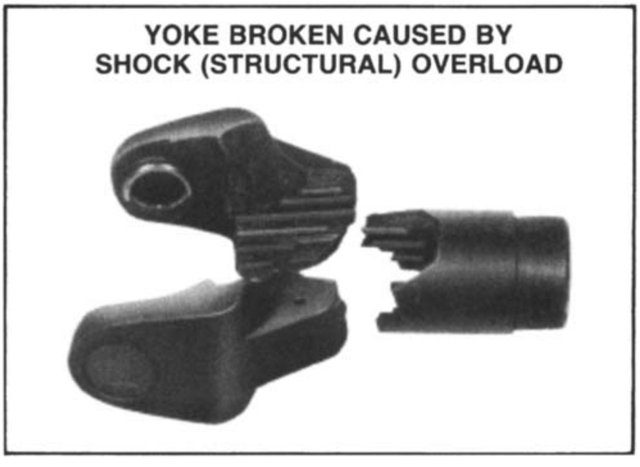


FIGURE 6.137 Yoke failure from overload. (Courtesy of Rockwell International.)

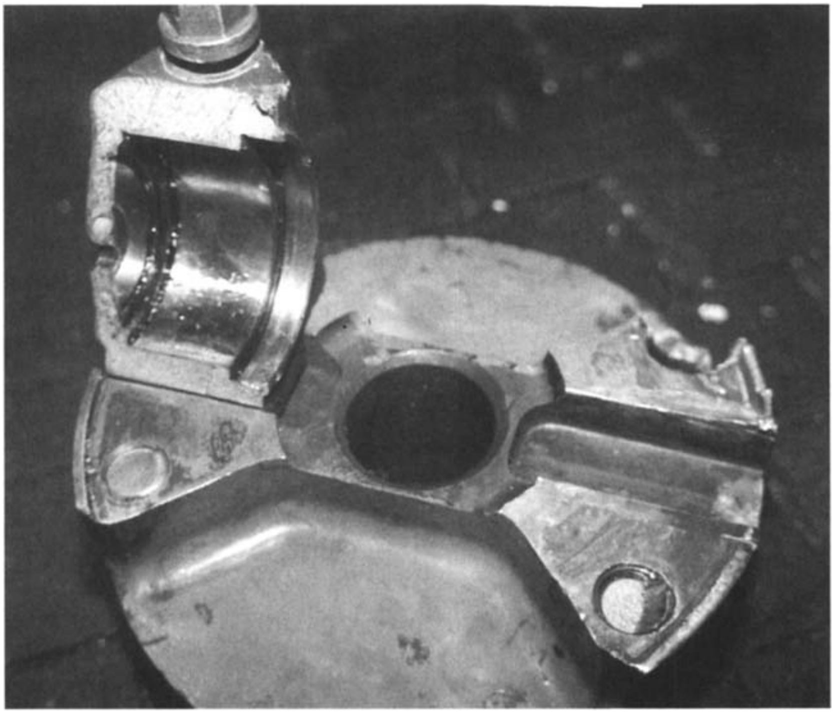


FIGURE 6.138 Block failure from overload.

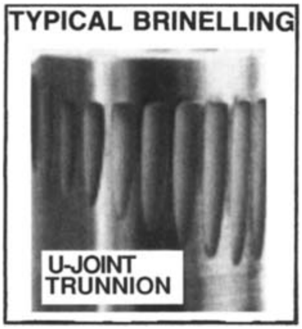


FIGURE 6.139 Trunion brinelling. (Courtesy of Rockwell International.)

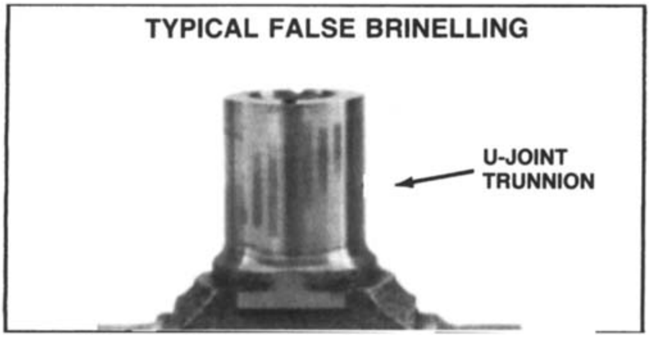


Figure 6.140 False brinelling of trunion. (Courtesy of Rockwell International.)

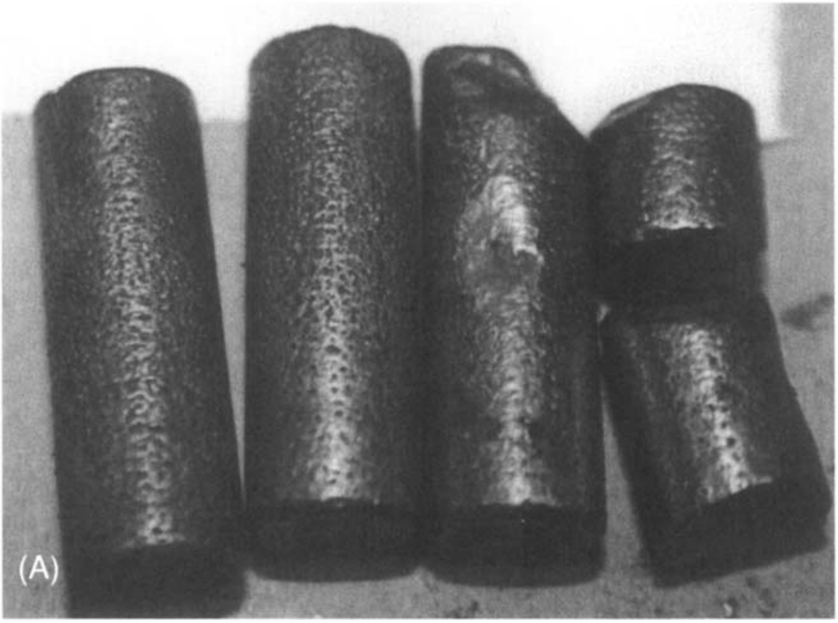


Figure 6.141 Pitting of universal. (A) Pitting of rollers. (B) Pitting of trunion.

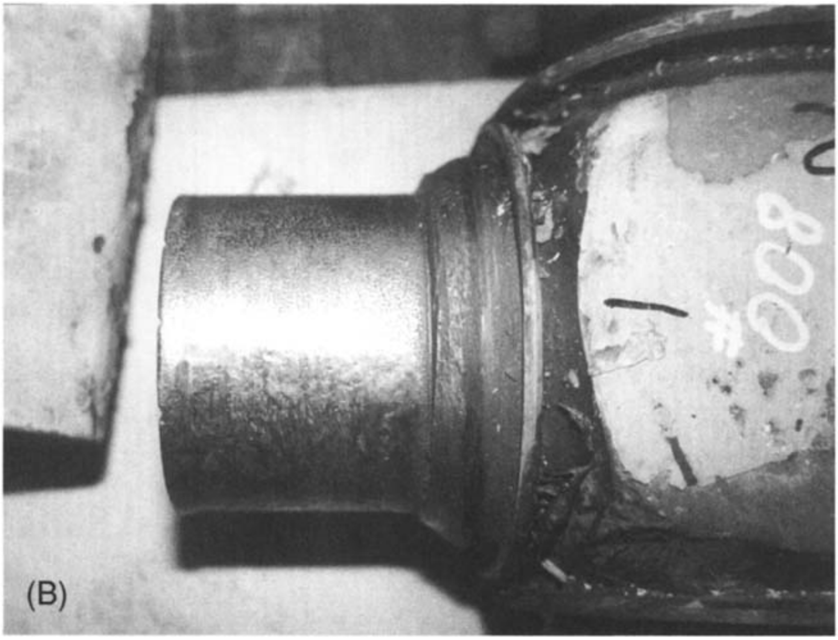


Figure 6.141 Continued.

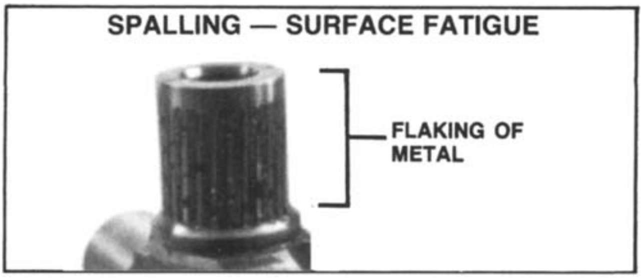


Figure 6.142 Spalling—surface fatigue. (Courtesy of Rockwell international.)

Complaint	Probable cause	Correction
	Universal joint center parts	
Fracture of cross or trunnion	Abnormally high loading	Check for abuse of vehicle (stuck, overloading)
	Capacity of universal joint too small	Check for maximum drive-line torque in lowest gear, compare this to capacity of universal joint series; if larger, substitute with larger series
	Operating angles excessive or uneven angles	Measure operating angles, if excessive, decrease to smaller angles
	Defective material	Replace with new parts
Fracture of bushing	Abnormally high loading	Check for abuse of vehicle (stuck, overloading)
	Capacity of universal joint	Check for maximum drive-line torque in lowest gear; compare this to capacity of universal joint series; if larger, substitute with larger series

FIGURE 6.143 Common problems, causes, and corrective measures for universal joints.

Complaint	Probable cause	Correction
	Universal joint center parts	
	Excessive and/or uneven operating angles	Measure operating angles; if excessive, decrease to smaller angles
	Defective material	Replace with new parts
	Trunnion appearance	
Light brinelling into trunnion	Normal bearing failure	If small area, no need to replace
	Lubrication breakdown	If low hours, use approved lubrication and frequent lubrication cycles
Spalling into trunnions	Lubrication breakdown	Increase frequency of lube cycle and/or use approved grease
	Dirt contamination	Increase frequency of lube cycle, and/or use approved grease
	Normal bearing fatigue	If high hours, replace with new parts
Galling on end of trunnion	Lubrication breakdown	Increase frequency of lube cycle, and/or use approved grease
	Excessive and/or uneven angles	Reduce angles

FIGURE 6.143 CONTINUED

	Unbalanced assembly	Check runout and balance of assembly
	Yokes: wing-type bushings or split eye	
Loose bolts	Dirt, foreign object, or material (paint, etc.) on mounting pad; look for fretting on mounting pads or drive tang	
Broken bolts	Over- or undertorqued bolts:	
	If there is no indication of fretting on mounting pad or bolt hole but there is fretting on the drive tang, the bolt broke	Surfaces must be free of foreign material and bushings fully seated before bolts are fully torqued
	If there is evidence of fretting, the bolt was loose	Surfaces must be free of foreign material and bushings fully seated before bolts are fully torqued
	Abnormally high angles can also cause bolts to break	Reduce angles to acceptable value

Figure 6.143 Continued

Complaint	Probable cause	Correction
Yokes: round bushing type		
Bushing extremely difficult to remove or replace	Hole in yoke distorted or normally close clearances; fretting corrosion and rust buildup	When removing bushing, use a penetrating oil and be careful not to distort ears on the yoke when removing on center cross Remove grease, rust, and dirt from mounting surfaces before assembly
New center parts will not flex when installed in yoke	Yoke ears distorted causing center parts to bind	Replace yoke

FIGURE 6.143 CONTINUED

7

Elastomeric Couplings

I. INTRODUCTION

As discussed earlier, flexible couplings have three basic functions:

1. To transmit power
2. To accommodate for misalignment
3. To compensate for end movement

Elastomeric couplings not only handle these three functions but also protect equipment from the damaging effects of shock loading and vibration through resilience.

There are two basic types of elastomeric couplings:

1. Elastomeric compression couplings
2. Elastomeric shear couplings

The elastomeric material is the resilient member, a rubber or synthetic rubberlike material. Figures 7.1 and 7.2 compare the two types of elas-

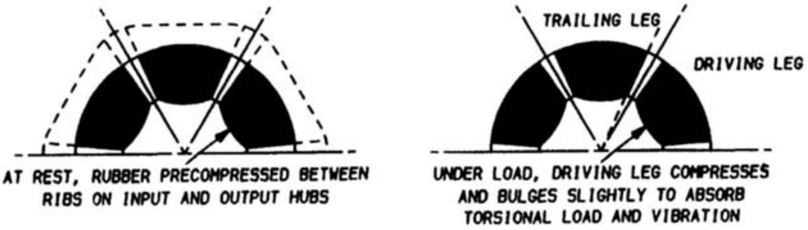

AT REST, RUBBER PRECOMPRESSED BETWEEN RIBS ON INPUT AND OUTPUT HUBS
TRAILING LEG
DRIVING LEG
UNDER LOAD, DRIVING LEG COMPRESSES AND BULGES SLIGHTLY TO ABSORB TORSIONAL LOAD AND VIBRATION

FIGURE 7.1 Elastomeric compression coupling.

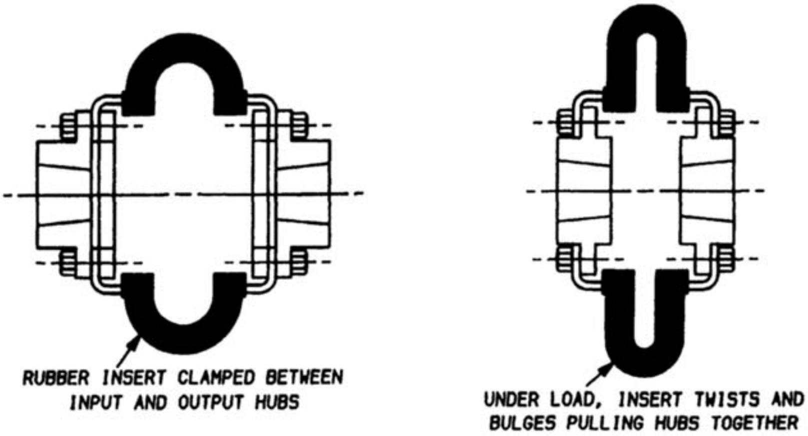

RUBBER INSERT CLAMPED BETWEEN INPUT AND OUTPUT HUBS
UNDER LOAD, INSERT TWISTS AND BULGES PULLING HUBS TOGETHER

FIGURE 7.2 Elastomeric shear coupling.

tomeric couplings: elastomeric compression coupling and elastomeric shear coupling, respectively. Elastomers can carry from 5 to 10 times more load in compression than in shear; however, they deflect more under shear than in compression. Thus couplings using elastomers in compression can transmit high loads and absorb minimal torsional vibration; those using elastomers in shear provide high torsional damping but cannot transmit high loads.

Figure 7.3 shows a typical curve of torque versus angular deflection, also known as a torsional stiffness curve, for various types of flexible couplings. Curve A is a typical torsional stiffness curve for a metallic element coupling (a disc coupling). Curve B is a typical torsional

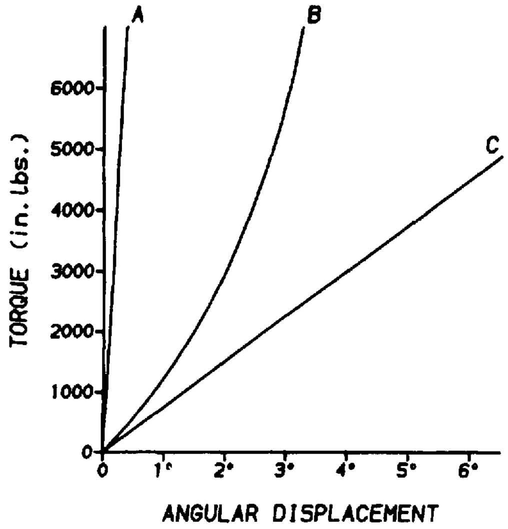


Figure 7.3 Torque versus angular windup.

stiffness curve for an elastomeric compression coupling. Curve C is for an elastomeric shear coupling.

A. Use of Elastomeric Couplings

Torsional vibration must be controlled to prevent equipment failure. An improperly selected coupling can cause component failure. A properly selected coupling, on the other hand, can prevent broken shafts, fretted splines, bearing wear or failure, damage to seals, and so on (see Figure 7.4).

When power transmission equipment components fail, users often blame the coupling rather than looking for the real cause. If more time and analysis were spent selecting the proper coupling for a system, fewer failures would occur. Equipment users may well assume that they have the best coupling for an application, never suspecting that it is the coupling that is causing the periodic failure of bearings, shafts, and other equipment components. They have thus come to expect these problems, scheduling premature overhauls and inspections that might well have been put off until a major equipment overhaul is really required.

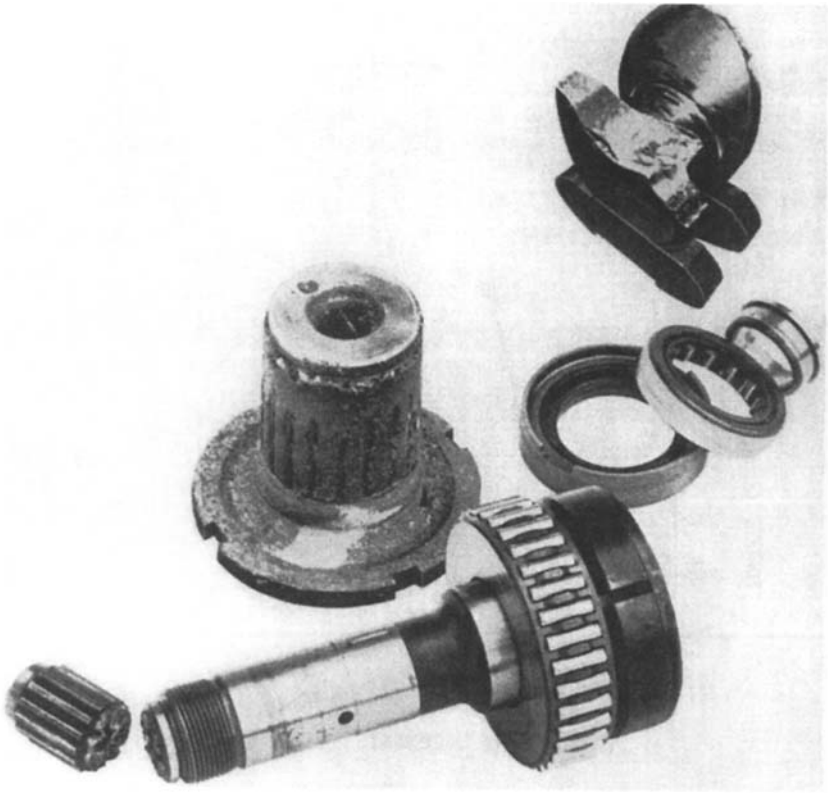


FIGURE 7.4 Failed and broken parts. (Courtesy of Lovejoy, Inc.)

The majority of the problems described are caused by vibration, and the phenomena of vibration are as complex as they are misunderstood. Described below is how vibration occurs and how its ill effects can be either eliminated or minimized In rotational equipment. Misalignment is the principal source of most vibration. In many years of industry experience, vibration due to misalignment has been found to be the largest cause of premature failure in rotating equipment. This type of vibration generates its own rotating force with exciting impulses that are related to the amount of misalignment and the speed of rotation.

Whether stationary or moving, every component and every system has what is called a "natural frequency." In rotational equipment, this frequency is a function of many factors, including mass, unbalance, and inertia moments. If a constant force is applied to a system, the system can operate at its natural frequency with no problem. But applied forces are rarely constant, particularly in engines and compressors. Thus a system involving the movement of many masses will have many natural frequencies.

As a system rotates, every cycle produces small impulses of energy, called *exciting impulses* because they can excite system vibration. Most systems are subject to some type of excitation, and with each impulse a cyclic force is applied to the system. The frequency of these cyclic forces is a function of the rotational speed of the equipment, and when this frequency is above or below the natural frequency of the system, the forces are absorbed by the system itself. When the frequency of the cyclic forces is in phase with the natural frequency, the forces create resonance at the natural frequency. When the magnitude of these forces exceeds acceptable limits, the system or a component within the system will vibrate and can cause premature failure of a component in the system.

The speed at which resonance occurs is known as the *critical speed*, the speed at which exciting impulses are in phase with the natural frequency. If this critical speed is within or too near the operating range of the system, the resulting vibration can cause problems.

Figure 7.5A illustrates an example of a problem. The vertical scale represents the equipment speed, and the horizontal scale merely converts these speeds into frequencies (or cycles per minute) by means of the diagonal line, known as the *critical order*. For any given speed, there is a corresponding frequency. The vertical line represents the natural frequency of this hypothetical system, and resonance will occur where the two lines cross, in this case at about 1700 rpm, which becomes the critical speed. The shaded area represents the operating range for the system, showing an idle speed of 1000 rpm and an operating speed of 1800 rpm. Since the operating speed is very close to the critical speed, the system is vulnerable to vibration, which can cause excessive wear and premature failure of many of the equipment's components.

Obviously, the problem outlined in Figure 7.5A could be corrected by lowering the operating speed of the equipment to 1600 rpm as shown in Figure 7.5B. This forces the critical speed outside the operating speed

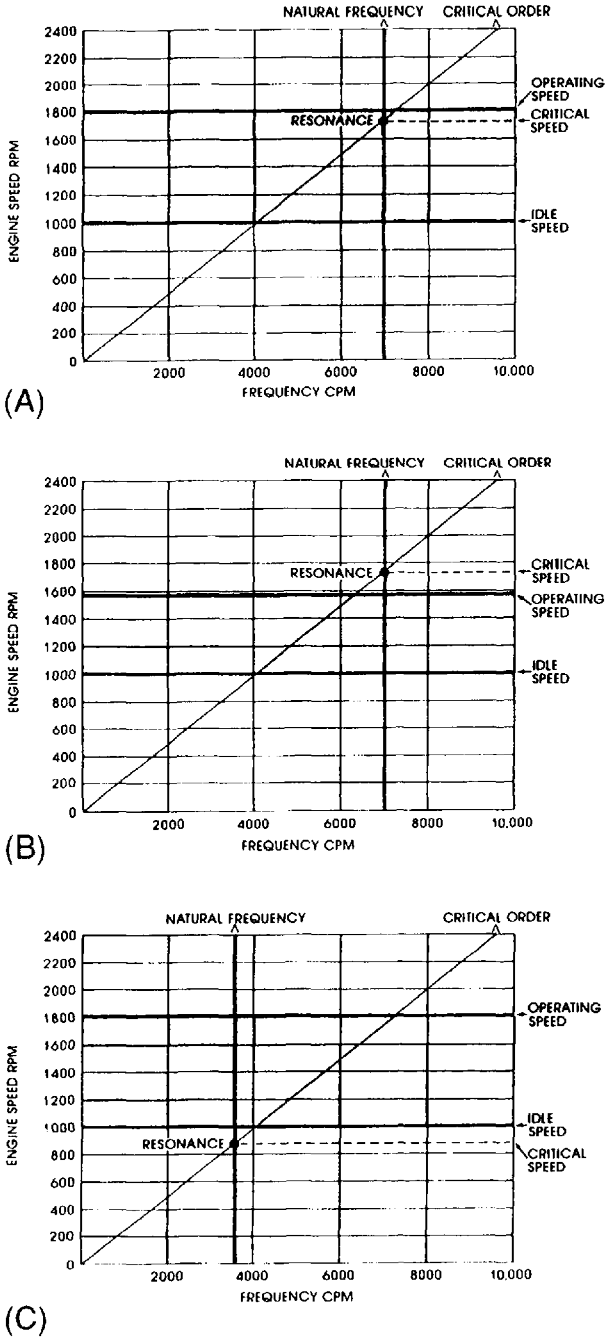


FIGURE 7.5 (A) Resonance too close to operating speed. (B) Lowering the operating speed. (C) Changing the inertia.

range of the system, thus avoiding the problem. However, this may not be practical, in which case the other remedy is to alter the critical speed of the system. This can be done by changing the weights (also the inertias) of the system, but that too may be impractical or impossible. Generally this would mean altering one of the system components; it may be too late or too expensive to exercise this option once the system has been designed and/or built. We have already determined that if it is impractical to alter the operating speed of the system; the other way to move the critical speed away from the operating speed is to change the natural frequency of the system itself as shown in Figure 7.5C. When it is impractical to do this by changing the system's inertia, it can be done with the addition of an elastomeric coupling.

B. Torsional Vibration

Since the primary function of elastomeric couplings is to control torsional vibration, one must understand torsional vibration to be able to apply an elastomeric coupling properly. The coupling manufacturer and/or the system supplier can usually do a system analysis to ensure that the selection is done properly. The following is a short dissertation on torsional vibration analysis.

Torsional vibration is continuous angular oscillatory motion. It can originate in a mechanical power transmission system from many sources: from variations in engine pressure, which can create peak torques; from motor pulsations; from unbalance forces; from fluctuating load torque requirements; or from torque reversals. When vibratory forces become large, they can produce damage, broken parts, excessive wear, and possibly noise in the rotating system.

Torsional problems sometimes appear to be complex and very difficult to understand. There are many reasons: Torsional problems usually occur less frequently than other types of problems, and the torsional equations contain radii and other factors that make the equations seem more complex.

One way to help understand torsional vibration is to recognize that there is a relationship between "translational" and "torsional" vibration. Both problems can be solved using the same basic formulas. Figure 7.6 shows the relationship between translational and torsional units. The

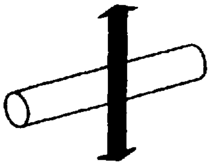
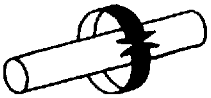
		
ELEMENT	TRANSLATION SYSTEM	TORSIONAL SYSTEM
MASS/INERTIA	$M=w/g$ (lb sec^2/in.)	$I=Mr_m^2=\frac{wr^2}{g}$ (lb in. sec^2)
SPRING	$K=$ lb /in.	$K=Kr_k^2$ (lb in./rad.)
DAMPER	$C=$ lb /in./sec	$C_R=Cr_c^2\frac{\text{in. lb sec}}{\text{rad.}}$
FORCE TORQUE	$F=$ lb	$T=Fr_t=$ lb in.
ACCELERATION	$A=\ddot{x}=$ in./sec^2	$\ddot{\theta}=\alpha=a/r=$ rad./sec^2
VELOCITY	$V=\dot{x}=$ in./sec	$\dot{\theta}=\omega=v/r=$ rad./sec
DISPLACEMENT	$D=x=$ in.	$\theta=$ in.$/r=$ radians

FIGURE 7.6 Relation between translation and torsional units.

only difference between the two types of vibration lies in the r or r^2 factors.

1. Terms

Mass (M) is a measure of resistance to rate of change or velocity. A body weighing (w) 1 lb on the earth will require a force of 1/386 lb to be accelerated at a rate of 1 in. per second per second.

Rotational inertia (I) is a measure of the resistance to change in rotation of a body about its center of rotation.

r is the radius at which the mass is concentrated with respect to its center of rotation. A torque of 1/386 in.-lb will cause a body with a weight of 1 lb and a radius of gyration $r = 1$ in to accelerate at a rate of 1 rad per second per second.

K is the spring rate and is a measure of *the resistance to displacement.* A spring that requires a force of 1 lb to displace it 1 in. has a spring rate of 1 lb/in. A torsional spring is measured in inch-pounds per radian. If $K = 1$ in.-lb/rad, then a torque of 1 in-lb will cause a rotation of 1 rad.

A *radian* is a length of arc on the circumference of a circle equal to the radius of the circle (see Figure 7.7A). It is approximately equal to 57.3° of arc.

The relationship between translation and torsional springs can perhaps be better understood by locating one end of a translation spring at point *A* in Figure 7.7B. Moving the end of the spring from *A* to *B*, a distance *r*, will require a force *K* (see Figure 7.7C). Since the force *K* is at a distance *r* from the center of rotation (O), the torque caused by the spring is $Kr \times r = Kr^2$. This torque Kr^2 was developed by moving the

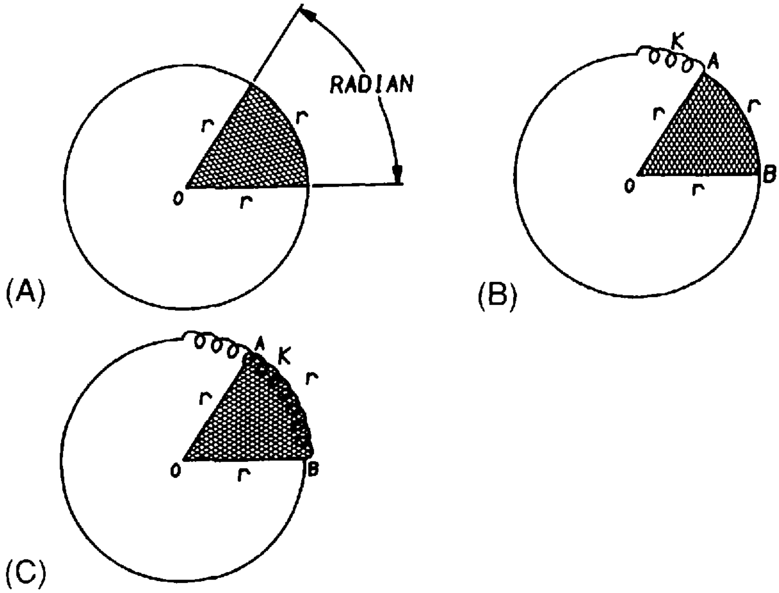


FIGURE 7.7 A radian is the angle subtended by an arc on the circumference of a circle whose length is equal to the radius of the circle. (B), (C) Spring movement.

spring a distance r, or 1 rad, along the circumference, so that the result is torsional stiffness in terms of inch-pounds per radian.

Damping C is a measure of resistance to velocity. A damper with a value of 1 lb per inch per second [1 lb/(in.-sec)] will exert a resisting force of 1 lb when motion across it is at the rate of 1 in./sec. Similarly, a torsional damper C_R will have a resisting torque of 1 in.-lb when motion across it is 1 rad/sec.

Remember that damping in elastomers is considered primarily when a system is vibrating at resonance. Usually, rotational systems are operating at a frequency away from resonance. Therefore, the frequency of a rotational system can usually be calculated without considering damping, and except at resonance damping has a negligible effect on transmissibility.

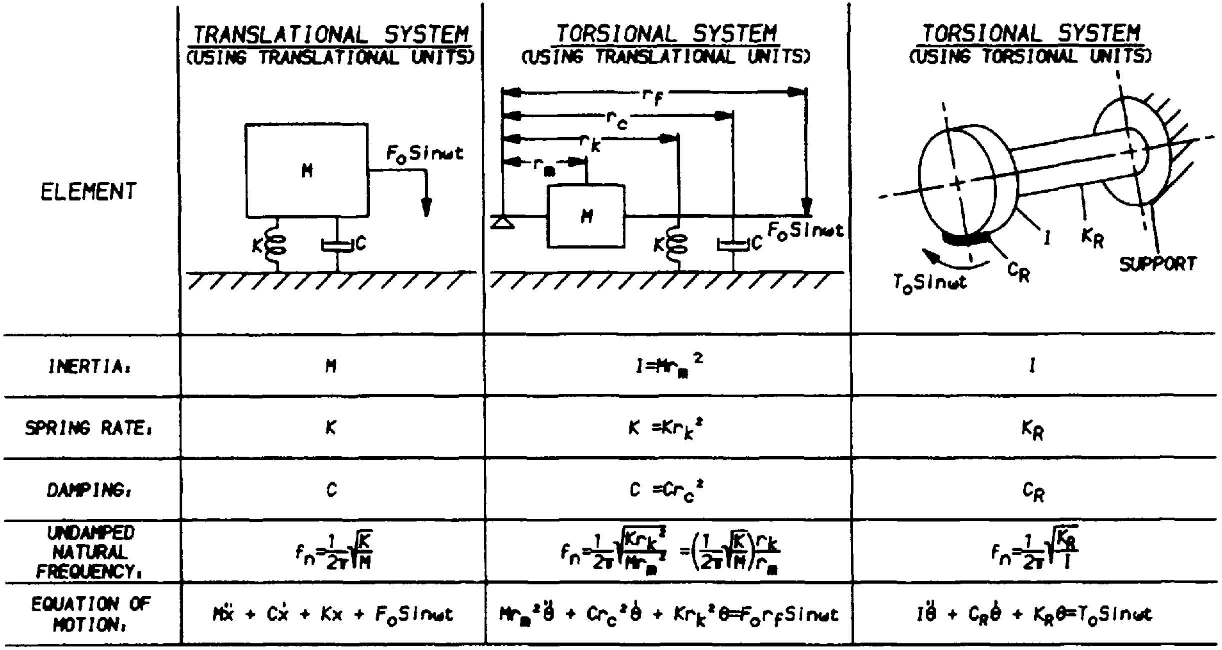
Figure 7.8 shows the same mass, spring, and damper in both translational and torsional systems. The elements have merely been rearranged in the torsional systems to rotate about a point O. This shows the similarity of the systems and the effects of the radius.

For the translational system, assume that the mass, spring, and damper are constrained to vertical motion. In torsional systems, motion about point O is limited to small values, so it is essentially vertical.

The formulas for inertia, rotational damping, and natural frequency in torsional systems indicate the effect of the radii—the distance from the center of the mass, spring, and damper—on the natural frequency of the system. The formula for the natural frequency of the torsional system in Figure 7.8 is identical to that of the translational system except that it is multiplied by the factor r_k/r_m. A torsional system rated in torsional values rather than in modified translational units is also shown in Figure 7.8.

2. System Torsional Design

Many designers have little difficulty in understanding single-degree-of-freedom torsional problems. In this case, the base or foundation can be assumed to have an infinite mass or inertia. Usually, however, vibratory systems are force-excited and consist of two or more inertias coupled with one or more springs. Solutions to problems in such a system are actually not much more difficult than those in single-degree-of-freedom systems.

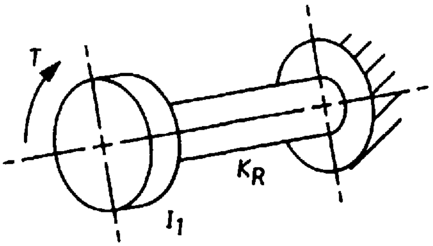


	TRANSLATIONAL SYSTEM (USING TRANSLATIONAL UNITS)	TORSIONAL SYSTEM (USING TRANSLATIONAL UNITS)	TORSIONAL SYSTEM (USING TORSIONAL UNITS)
ELEMENT			
INERTIA:	M	$I = Mr_m^2$	I
SPRING RATE:	K	$K = Kr_k^2$	K_R
DAMPING:	C	$C = Cr_c^2$	C_R
UNDAMPED NATURAL FREQUENCY:	$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$	$f_n = \frac{1}{2\pi}\sqrt{\frac{Kr_k^2}{Mr_m^2}} = \left(\frac{1}{2\pi}\sqrt{\frac{K}{M}}\right)\frac{r_k}{r_m}$	$f_n = \frac{1}{2\pi}\sqrt{\frac{K_R}{I}}$
EQUATION OF MOTION:	$M\ddot{x} + C\dot{x} + Kx + F_o \sin \omega t$	$Mr_m^2\ddot{\theta} + Cr_c^2\dot{\theta} + Kr_k^2\theta = F_o r_f \sin \omega t$	$I\ddot{\theta} + C_R\dot{\theta} + K_R\theta = T_o \sin \omega t$

FIGURE 7.8 Comparison of translation and torsional systems.

The first fact to consider is that most multi-degree-of-freedom torsional systems have two inertias and one spring or can readily by reduced to an equivalent two-inertia, one-spring system with little error. If a more complex situation is presented, a complete analysis can be carried out by referring to the Holzer analysis, given in many books on torsional vibration.

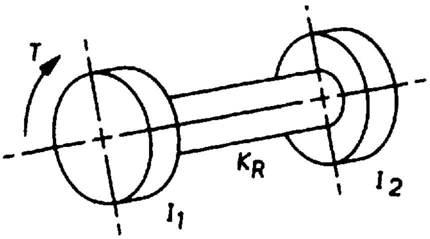
To compare a single-degree-of-freedom system to two-inertia, one-spring systems, refer to Figures 7.9 and 7.10. Figure 7.9 is a single-degree-of-freedom system with an inertia I_1 connected to a spring K_R attached to a rigid support. Figure 7.10 represents a two-degree-of-freedom system with a force imposed on inertia I_1, through a spring K_R, to inertia I_2, which is also free to rotate. The formulas below the figures give the calculations of natural frequency and transmissibility for each of these systems. Absolute transmissibility, T_{abs}, represents the ratio of the torque transmitted through the spring, K_R, to the support (Figure 7.9) or to the driven inertia (Figure 7.10) I_2, to the torque applied to the driving inertia I_1.



$$f_n = \frac{60}{2\pi}\sqrt{\frac{K_R}{I_1}} \text{ cycles/min}$$

$$T_{abs} = \frac{1}{1-\left(\frac{f}{f_n}\right)^2}$$

FIGURE 7.9 Single-degree-of-freedom system.



$$f_n = \frac{60}{2\pi}\sqrt{\frac{K_R}{I_{Eff.}}} \text{ cycles/min}$$

$$T_{abs} = \frac{I_2}{I_1+I_2} \times \frac{1}{1-\left(\frac{f}{f_n}\right)^2}$$

WHERE $I_{Eff} = \frac{I_1 I_2}{I_1+I_2}$

Figure 7.10 Two-inertia, one-spring system.

3. Transmissibility

To understand the relationship between single-degree-of-freedom and two-inertia, one-spring systems, examine Figure 7.11. Curve A represents a single-degree-of-freedom system. It shows how transmissibility increases from 1:1 at very low frequencies to infinity when the disturbing frequency equals the natural frequency. Then it moves into the isolation range, where transmissibility is less than 1 when the ratio of the disturbing frequency to the natural frequency, f/f_n, is more than 2 or 1.41.

Curve B shows the torque transmissibility in an application where the inertia of the driven part of the system is approximately one-tenth that of the driving part. In this case, the transmissibility is 0.1 at very low frequencies. It then increases in the resonance range until the disturbing frequency is equal to the natural frequency, then rapidly decreases in the isolation range.

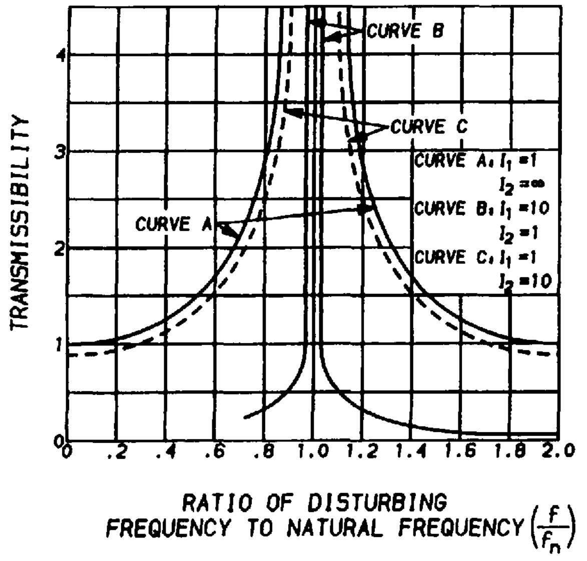


Figure 7.11 Comparative transmissibility curves.

Referring to the transmissibility formulas in Figures 7.9 and 7.10 and to curves A and B in Figure 7.11, it is obvious that the transmissibility of the two-inertia system is approximately one-tenth of the transmissibility of the single-degree-of-freedom system throughout the frequency range, except at resonance.

Curve C of Figure 7.11 shows how a two-inertia, one-spring system is approximately equal to a single-degree-of-freedom system when the driven inertia is 10 or more times the magnitude of the driving inertia. When I_1 is small compared to I_2, the fraction $I_2/(I_1 + I_2)$ of a two-inertia, one-spring transmissibility formula approaches that of a single-degree-of-freedom system. Transmissibility based on a single-degree-of-freedom formula will result in a conservative value.

The next factor to be considered in the calculation of torsional systems is gear or belt drives with different speeds between various parts of the system. Where such differences exist, inertias must be corrected for their effective inertia at the coupling speed before frequencies and transmissibilities can be calculated. This correction is made by multiplying the inertia of an element rotating at a speed different from the coupling speed by the factor $\mathrm{rpm}_l^2/\mathrm{rpm}_c^2$, where rpm_l is the speed of the element and rmp_c is the speed of the coupling.

C. Failure Modes for Elastomeric Couplings

When an elastomeric coupling melts or breaks apart and spins off rubber, it is obvious that something went wrong. Failures of elastomeric couplings may be the symptoms of a severe mechanical problem within the system. No matter how an elastomeric coupling fails, it can usually tell us why it failed. There are five common causes of failure:

1. Misalignment
2. High thrust loads
3. High torsional loads
4. Improper selection, operation, or installation and/or maintenance
5. Hostile environment

Figure 7.12 is a list of the ways in which elastomeric couplings typically fail.

1. Misalignment of Excessive Misalignment

Elastomeric couplings are designed to accept a certain amount of misalignment; when this is exceeded, identifiable symptoms are usually recognizable. When, for example, a jaw coupling generates an audible noise, the cause is usually wearing of the rubber spider. Bonded-tire elastomeric couplings often suffer bond failure between the tire and the metallic hub because of the flexing required by excessive misalignment.

In general, when the flexing member shows worn lugs, teeth, spiders, pins, bushings, and so on, excessive angular and/or parallel misalignment is the usual cause. Excessive misalignment can also cause failure of shafts, bearings, and other equipment parts due to excessive loads trying to flex the coupling.

Failure mode	Probable cause	Corrective action
Worn flexing element or shaft bushings Shaft bearing failure High-pitched or staccato noise	Excessive shaft misalignment	Realign coupling and shafts to meet specified tolerances
Ruptured elastomeric flexing element Sheared hub pins or teeth Loose hubs on shaft, sheared keys	Torsional shock overload	Find and eliminate cause of overload Use larger coupling
Fatigue of flexing element Overheated elastomeric tire or sleeve Fatigue of hub pins or discs Worn gear teeth Staccato or clacking noise Loos hubs on shaft, keyseat wallow	Torsional vibration Excessive starts and stops High peak-to-peak torsional overload	Use larger coupling Use larger coupling Add flywheel to hub
Shaft bearing failures High-pitched wine Motor thrust bearing failure	Lubricant failure	Replace or rebuild coupling
Swollen or cracked elastomeric flexing member Lubricant failure Severe hub corrosion	Chemical attack	Use more chemically resistant flexing member or hub Coat hubs Coat hubs
Distorted or deteriorated elastomeric flexing member Lubricant failure	Excessive heat	Use more heat-resistant flexing member or lubricant
Shattered flexing member Lubricant failure	Low temperature (below 0°F)	Use special low-temperature rubber compounds and lubricants

FIGURE 7.12 Typical elastomeric coupling failures.

2. High Thrust Loads

Flexible couplings used on equipment such as agitators, mixers, and propeller shafts often must carry axial thrust loads. Excessive "inward" thrust can badly distort shear couplings and place high stresses on them. Reverse or outward thrust cannot be carried by couplings that can be pulled apart. Such loads pull the coupling out of the mesh and heavily load the ends of the coupling members (see Figure 7.13).

3. High Torsional Loads

Although flexible couplings are built to carry torsional loads, a sudden impact can cause them to fail. Shear couplings rupture under torsional impact to protect the coupled machinery. In failure, these couplings develop a 45° tear or series of tears over the insert.

Torsional peak loads from reciprocating machinery also fatigue capscrews or fasteners that secure elastomeric bushings, compression members, and flexible elements to hubs. In some instances, these loads can build up to as much as 7 times the normal operating load. To prevent premature failures on rotating equipment, a torsional vibration analysis should be run as part of the coupling selection process. The rotational inertia of the two coupled machines, operating speed, and torsional stiffness of the coupling are the main inputs to the analysis. A tor-

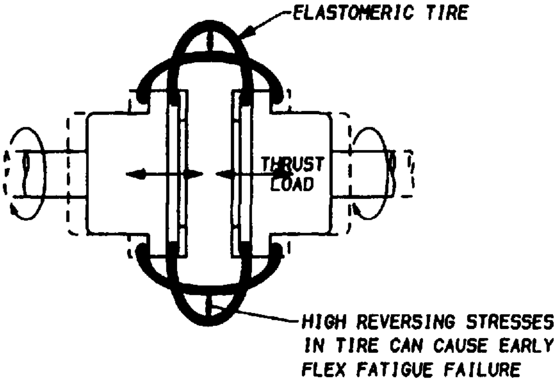

ELASTOMERIC TIRE
THRUST LOAD
HIGH REVERSING STRESSES IN TIRE CAN CAUSE EARLY FLEX FATIGUE FAILURE

FIGURE 7.13 Thrust-load distortion of coupling members.

sional analysis study should always be rechecked if the type and size of the coupling is changed or replaced.

Manufacturers of reciprocating compressors commonly use torsionally stiff couplings to connect the compressor to the driver. In effect, the driver rotor becomes a flywheel that smooths out the torsional pulses inherent in these drives. A torsionally soft coupling isolates the driver and the compressor, forcing the coupling to absorb both high peak torques and any torque reversals. Elastomeric tire or insert-type couplings fatigue when subjected to high peak torsional loads, while jaw and pin-and-bushing couplings emit a high-frequency noise. A flywheel, then, is a necessity when torsionally soft couplings are used on reciprocating machines.

If torsional overloads are the cause of premature coupling failure, one of the following system changes should be made:

1. Eliminate the cause of the overload by changing a system component.
2. Reduce the magnitude of the torsional vibrations by detuning the system; that is, use a coupling that allows the system to operate at noncritical frequencies.
3. Increase the coupling size.

4. Improper Selection, Operation, Installation, and Maintenance

The three major causes of coupling failure are improper sizing, unexpected operation conditions, and lack of periodic installation and maintenance. There are no industry standard service factors for elastomeric couplings. Thus, one manufacturer uses a 1.5 service factor for centrifugal pumps, whereas another may recommend using a 1.25. *It is important to use the factors that are recommended by the specific coupling manufacturer.* Also, some coupling ratings are based on constant torque (horsepower varies directly with speed), whereas others are based on an exponential relationship between power and speed. Comparisons among competitive couplings should be made as discussed in Chapter 3.

Frequently, an oversized elastomeric coupling is specified in the hope that it will last longer. However, this may not be true, and it increases the radial loads on the shafts if misalignment exists. Also, if the load transmitted is too low, the oversized elastomeric coupling may actually wear out much faster. Occasionally, a coupling must be oversized to fit the shafts. In these cases, care must be taken to align and install the coupling more accurately than usual.

Speed limits should be strictly adhered to on elastomeric couplings used in high-speed operations. This type of operation generates centrifugal forces that can place high stresses on the flexible coupling components, particularly at the hub bore. Stress concentrates around drilled and tapped holes used to secure the hubs to the shaft. Tire-type elastomeric couplings can "balloon" and eventually rupture at high speeds. Compression-type elastomeric type couplings may whirl or vibrate laterally. Many of these couplings may need some type of lateral support for use at high speeds (internal bearing; see Figure 7.14).

Stated limits can sometimes be increased if high-strength metal hubs and special inserts or flexing members are used. However, hubs and other components may have to be balanced to help prevent vibration. Coupling manufacturers should be consulted about when they require that their couplings be balanced. As always, the couplings should be handled carefully during installation and removal. Rough handling can cause internal fractures in the hub casing, which may then fly apart on startup. Extreme care should also be taken when installing elastomeric flex elements to avoid tearing or breaking them. Always follow the coupling manufacturer's installation and maintenance instructions.

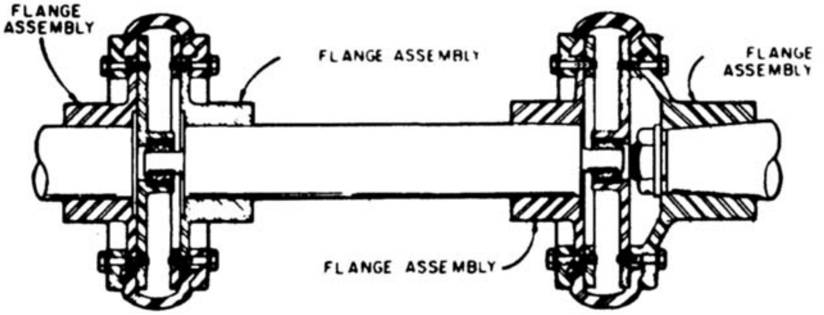


FIGURE 7.14 Tandem rubber tire coupling. (Courtesy of Dodge Division of Reliance Electric.)

Resistance to:	Neoprene[a]	EPDM[a]	Urethane[a]	Hytrel[a]
Acetone	2	1	3	2
Ammonium hydroxide solutions	1(158°F)	1	1	—
Benzene	3	3	3(158°F)	2
Butane	1	2	1	1
Carbon tetrachloride	3	3	3(122°F)	3
Chloroform	3	3	3	3
Ethyl alcohol	1(158°F)	1	3	1
Ethylene glycol	1(158°F)	1	2	1
Fuel oil	1	—	2	1
Gasoline	2	2–3	2	1
Glycerin	1(158°F)	—	1	1
Hydraulic oils	1	—	2	1
Hydrochloric acid (20%)	1	—	2	2
Isopropyl	1	—	3	1
Kerosene	2	—	3	—
Lubricating oils	2(158°F)	—	2	1
Methyl alcohol	1(158°F)	—	3	1
Mineral oil	1	—	1	1
Naphtha	3	3	2	1
Nitric acid (10%)	2	—	3	3
Nitrobenzene	3	1	3	3
Phenol	2	—	3	3
Soap solutions	1(158°F)	—	1	1
Sodium hydroxide (20%)	1	1	—	1
Sulfuric acid (50–80%)	2–3	—	3	3
Toluene	3	3	3(122°F)	2
Trichlorethylene	3	—	3	3
Turpentine	3	3	3	—
Water	1(212°F)	1(158°F)	1(122°F)	1(158°F)
Xylene	—	3	3	2

[a] 1, little or no effect; 2, minor to moderate effect; 3, severe effect. Where temperatures are noted, resistance drops off quickly above the stated temperature.

FIGURE 7.15 Chemical resistance of flexible coupling material.

5. Hostile Environment

Elastomeric couplings are often used in corrosive and abrasive environments and at widely varying temperatures (see Figure 7.15). On food processing machinery, for example, many couplings must withstand daily washing with hot water as well as resisting corrosion and deterioration from salt spray. These environments can cause corrosion of metallic elements and/or deterioration of elastomeric elements. In addition, sunlight quickly ages natural rubber flexing members, resulting in dry rot failure. Fortunately, heat generated when the coupling operates at the critical frequency often warms the flexible member enough to protect the system.

II. ELASTOMERIC COMPRESSION COUPLINGS

A. Types and Uses of Compression Couplings

There are three basic types of elastomeric compression couplings:

1. The block/jaw coupling
2. The pin-and-bushing coupling
3. The donut or ring (lug/clamped or bushed) coupling

1. Block/Jaw Couplings

The block/jaw type of elastomeric coupling has an elastomer in compression. One-piece flexible members (see Figure 7.16) are generally used on low-to-medium horsepower applications. On high-horsepower applications, individual elastomeric elements can be used (see Figure 7.17A). Elastomers in compression have a fail-safe feature. The rubber blocks are installed in cavities formed by driving and driven jaws (blades). If the elastomer fails, the coupling may run for some time on the metal jaws (bladed). Flexible elements can be composed of many types of elastomeric materials such as rubber and urethane. The properties (hardness, resilience, etc.) can be varied to suit the required loads. The torsional stiffness, torque capacity, and overall dimensions can be altered by increasing or decreasing the number of jaws, the jaw width, and/or the shape of the jaw. Some of these couplings are used primarily to accommodate misalignment and transmit power. In applications where small and medium-sized equipment employs these couplings one needs to be careful, since sometimes the couplings are used to absorb energy and dampen loads, particularly when the equipment is engine-driven. Also, sometimes they are used because they are torsionally soft and are used to tune a system out of a torsional critical mode.

On large critical equipment (such as synchronous motor-driven equipment), use individual elastomeric block type couplings (Figure

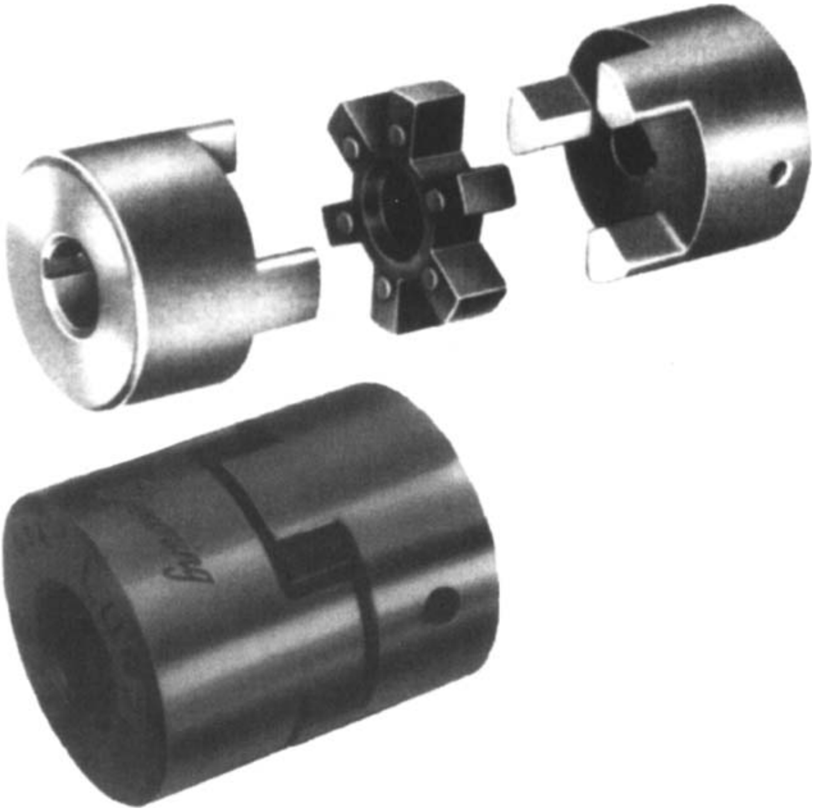


FIGURE 7.16 Jaw coupling. (Courtesy of Browning, Emerson Power Transmission Corporation.)

7.17A) to reduce vibratory torques or to torsionally "tube" a system (using its torsional "softness." Since block elastomers are almost incompressible, the cavities are constructed so as to allow proper deformation of the blocks and are completely filled only under conditions of extreme overload, combining high load-carrying capacity with maximum resiliency. These blocks are usually precompressed into their respective cavities. As torque is applied, alternate driving blocks flex under the increased compressive load and tend to assume the shape of the enclosing cavities. The trailing blocks are relieved, but never com-

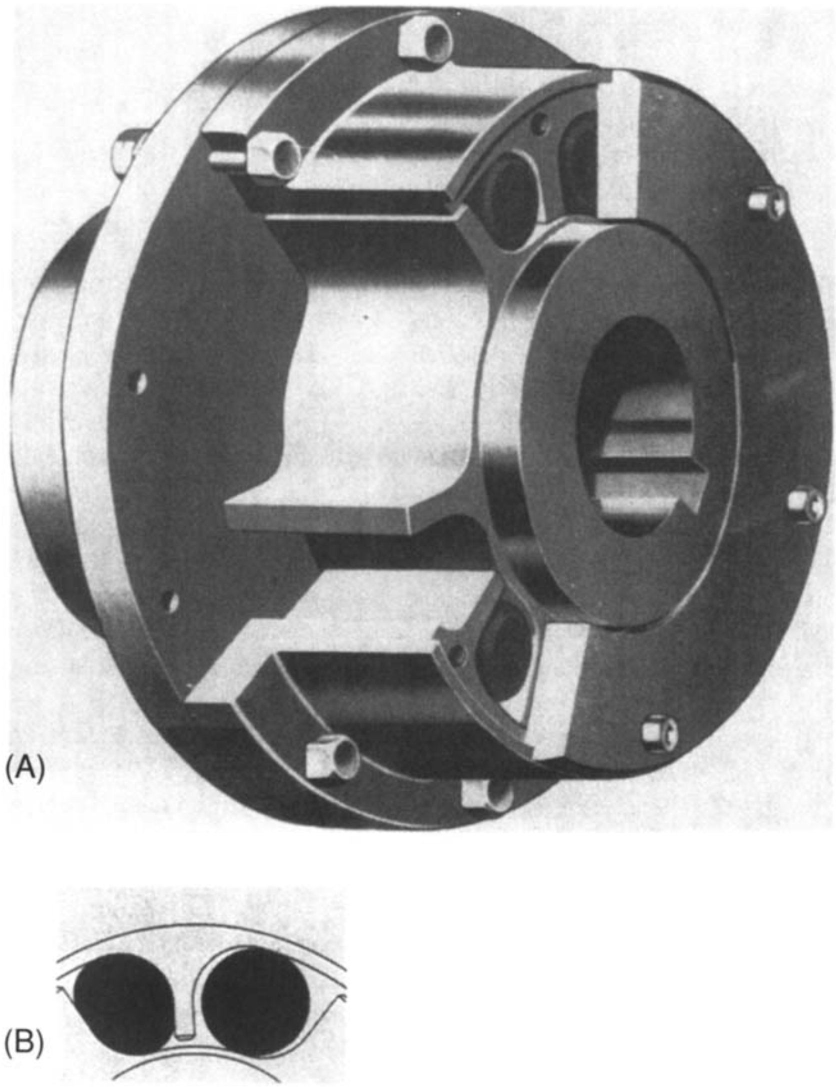


Figure 7.17 (A) Block elastomeric coupling. (B) Loading of elastomeric blocks. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

pletely, and tend (but not quite) to resume their original molded shape (see Figure 7.17B). The alternating loaded or driving blocks tend to fill cavities (as shown by the block on the left in Figure 7.17B) as greater deformation occurs. The alternate unloaded or trailing blocks (the block shown on the right), being precompressed, effect a smooth transition upon reversal of torque with complete absence of backlash.

2. Pin-and-Bushing Couplings

The elastomeric bushing (Figure 7.18A) has been proved in countless applications in power transmissions over many years. It is a universal coupling without working parts in the accepted sense. All relative movement is accommodated by the controlled internal displacement of compressed cylindrical rubber blocks having specially shaped end profiles. The bore of the bushing is steel bonded into the rubber during vulcanization.

Preloading of the bushing is illustrated in Figure 7.18B, where *A* is the diameter of the biscuit in the free state and *B* is the diameter of the biscuit after insertion into the housing, showing the biscuit in a preloaded condition. This preloaded condition and the special shape of the biscuit accommodate any movement through controlled internal displacement of the elastomeric material.

Axial displacement resulting from thrust loads is illustrated in Figure 7.18C, where *A* indicates the position of the bushing prior to imposition of thrust load, and *B* indicates the position of the bushing after a thrust load has been imposed. The flow of elastomeric material permits free end float without creating undue thrust loads on driveshaft bearings.

Angular deflection is illustrated in Figure 7.18D, where *A* indicates the centerline of the bushing before angular deflection, and *B*, the displacement of the elastomeric material, as indicated by arrows, compensates for angular misalignment of the connected shafts.

Figure 7.18E illustrates torsional deflection resulting from torque loads and torsional vibration. Here *A* is the centerline of the bushing before the application of a horizontal load. The imposition of a torque load increases pressure in the direction of the load and reduces pressure in the opposite direction, moving the centerline to position *B*. Because of the initial preloaded condition, the elastomeric bushing is still under compression throughout its volume, even at maximum torque load.

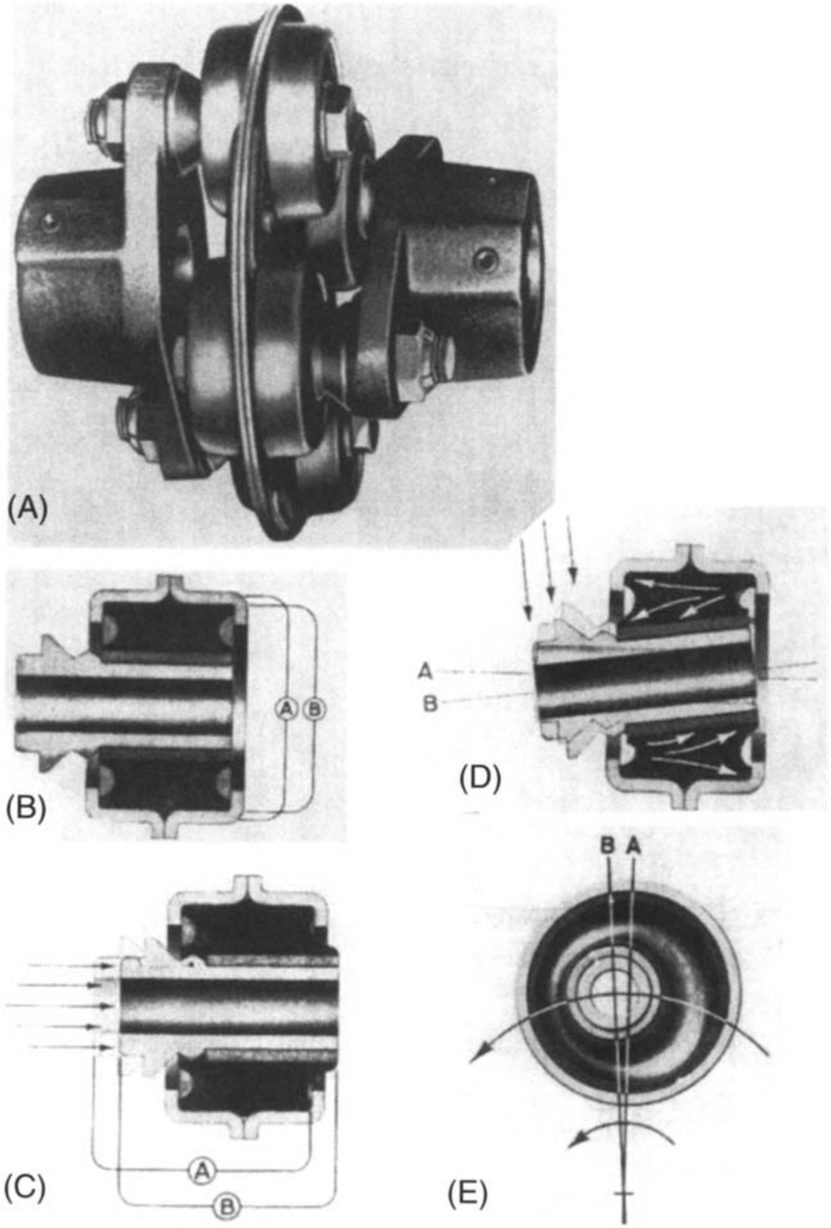


FIGURE 7.18 (A) Typical pin-and-bushing coupling. (B)–(E) Loading bushings. (Courtesy of Morse Industries, Emerson Power Transmission Corporation.)

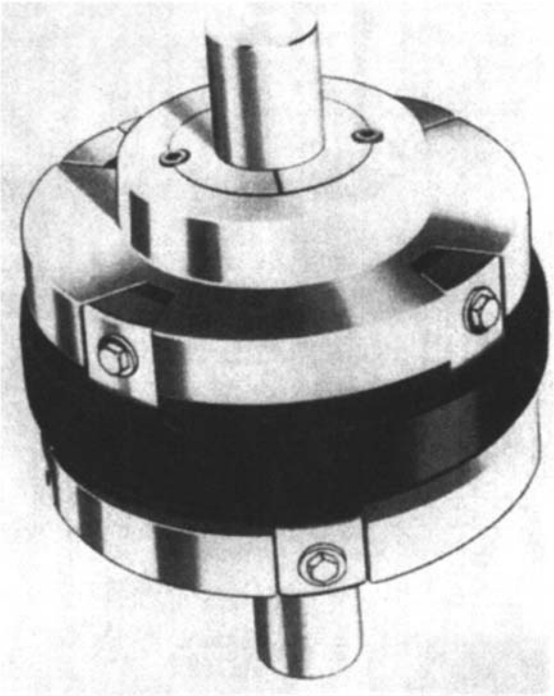


FIGURE 7.19 Elastomeric donut coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

3. Donut or Ring Couplings

a. Elastomeric Donut Lug, Clamped or Restrained The coupling shown in Figure 7.19 has an elastomeric donut in a precompressed state because of the dimensional tolerances of its mating components. The donut is forced to a diameter smaller than its natural diameter when the capscrews are drawn up. This concept places all legs of the donut in compression before a load is applied. In operation, the trailing legs stretch, but not Into tension. Absence of tension benefits the rubber element by minimizing the ozone attack that usually accompanies rubber in tension. This type of coupling is used on small-horsepower equipment (generally under 100 hp).

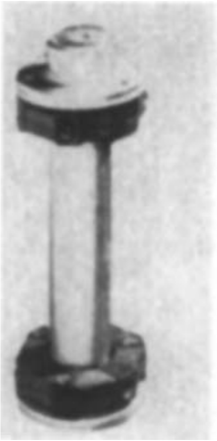


Figure 7.20 Elastomeric coupling with lugs. (Courtesy of Lovejoy, Inc.)

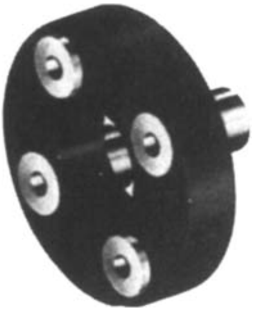


Figure 7.21 Elastomeric donut with bushings. (Courtesy of the Lord Corporation.)

b. Elastomeric Donut (Lug or Bushing) Type This coupling (Figures 7.20 and 7.21) was specially developed for connecting high-inertia drives such as diesel engines to low-inertia driven members such as hydrostatic pumps. The material of the elastomeric element provides a torsionally stiff coupling. High permissible rotational speeds are attained, and minor misalignments are accommodated. The hardware design allows this coupling to connect directly to flywheels or to connect using simple flywheel adapter plates. This type of coupling is used on small-to-medium-horsepower equipment, generally under 500 hp.

III. ELASTOMERIC SHEAR COUPLINGS

A. Types and Uses of Shear Couplings

There are three basic types of elastomeric shear couplings:

1. The tire coupling
   a. Urethane tire coupling
   b. Corded tire coupling
2. The bonded coupling
3. The donut coupling

1. The Tire Coupling

a. Urethane Tire Coupling. This coupling (Figure 7.22) has its elastomeric element in shear and is made of urethane. The tire is split to enable easy assembly without removing the hub. Urethane couplings

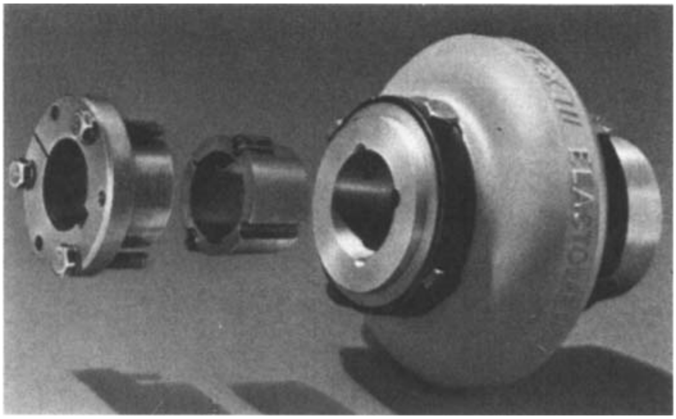


FIGURE 7.22 Urethane tire coupling. (Courtesy of Rexnord Elastomeric Products Operation.)

offer a high degree of flexibility. When the coupling fails, usually only the elastomeric element is replaced. This type of coupling is used on small-horsepower equipment, generally under 100 hp.

b. Elastomeric Tire Coupling This coupling (Figure 7.23) has the reinforced flexing element at the outermost radius of the configuration, thereby permitting a small ratio of overall length to torque capacity. The flexing element is subjected to shear. Internal reinforcement and external clamping of the tire increases the torque capacity and overall stiffness over that of a comparable unclamped and unreinforced shear unit of the same package size. This type of coupling is used on small-horsepower equipment, generally under 100 hp.

Three-bar linkage principle. The shape factor or cross-sectional profile of the elastic member connecting the hubs is very important if the coupling is to maintain maximum flexibility and a cushioning effect under misaligned conditions. Figure 7.24 shows a simple pinned three-bar linkage that will readily accept movement or misalignment. Figure 7.24A shows the normal position of the linkage. Figure 7.24B shows the action of the bars with the right-hand bar displaced horizontally along the reference baseline. This motion simulates the end float or axial misalignment in a flexible coupling. Figure 7.24C shows the relative position of the bars when the right-hand bar is displaced in a vertical plane above or below the baseline, simulating the parallel misalignment most commonly found in drive system couplings. Angular misalignment in the plane of the paper is illustrated in Figure 7.24D.

If we apply this principle to a coupling, replacing the three-pinned bars with a continuous elastic material such as rubber, the motions of misalignment will be approximately the same. Figures 7.24E–G show the translation of the pinned three-bar linkage principle to an arched elastomeric material connected to the hubs of a coupling. The similarity of misalignment capabilities is evident. The elastic material will bend or deform to allow displacement of the coupling hubs for misalignment.

Two types of elastomeric couplings. Figure 7.25 is a cutaway view of a flexible coupling with an arched profile. A positive method of securing each vertical leg of the connecting member between the shaft hub or flange and an internal clamping ring makes it impossible for the

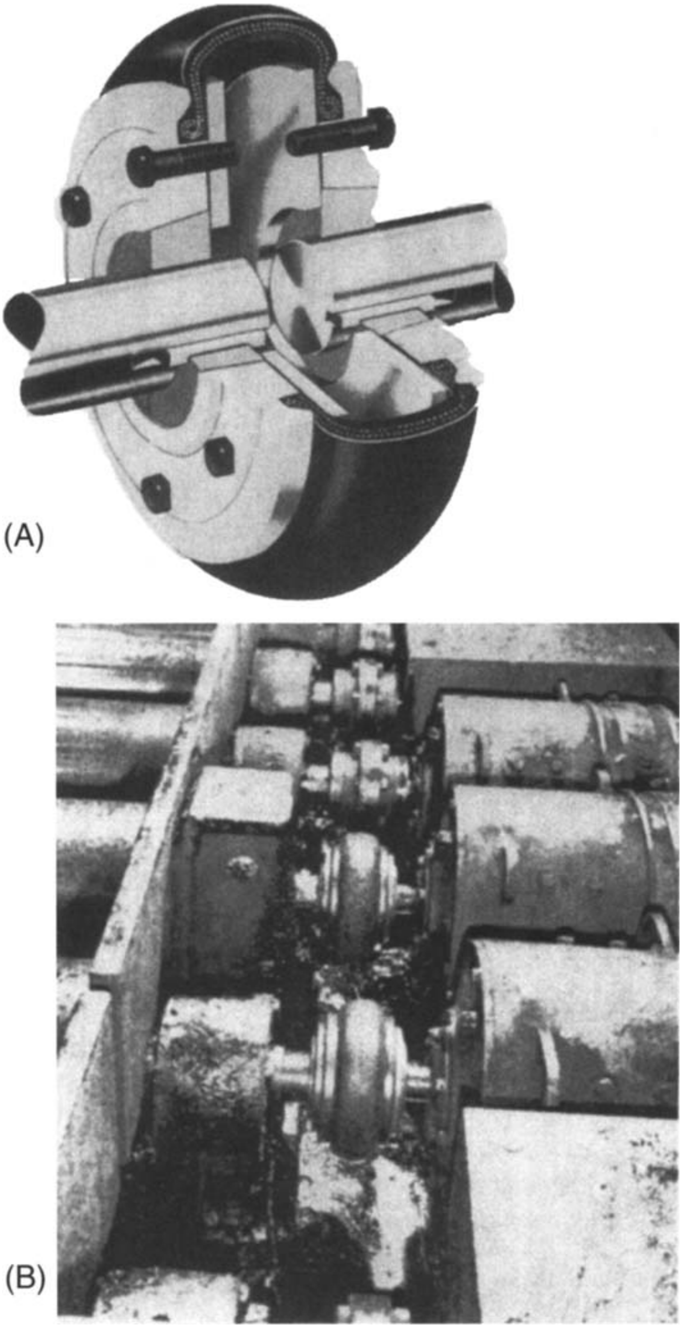


FIGURE 7.23 (A) Elastomeric tire coupling. (B) Elastomeric tire coupling for runout table (Courtesy of Dayco Corporation.)

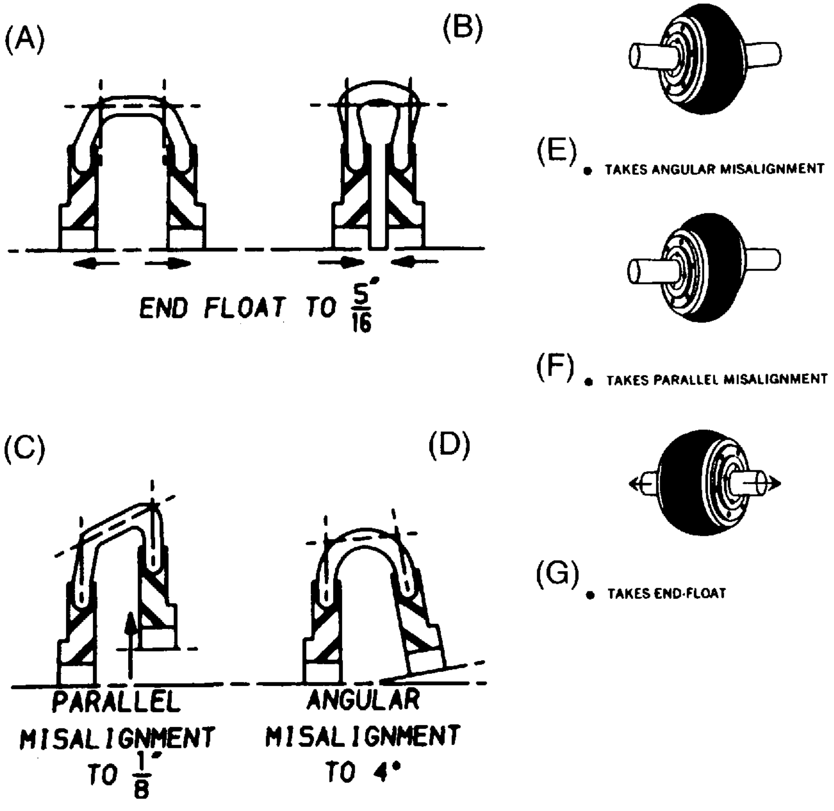

(A)
(B)
END FLOAT TO 5/16"
(E) • TAKES ANGULAR MISALIGNMENT
(F) • TAKES PARALLEL MISALIGNMENT
(C)
(D)
(G) • TAKES END-FLOAT
PARALLEL MISALIGNMENT TO 1/8"
ANGULAR MISALIGNMENT TO 4°

Figure 7.24 (A)–(D) Three-bar linkage principle. (E)–(G) Misalignment motion of a tire coupling. (Courtesy of Dayco Corporation.)

soft resilient member to rub or slide against metal to cause early failure during misaligned operation. There are no projections from either shaft flange to interfere with or bind the coupling during severe misalignment. The flexible element, shaped much like an automobile tire, is split radially at one location through the cross section. This allows it to be installed or replaced without moving or distributing the drive and driven

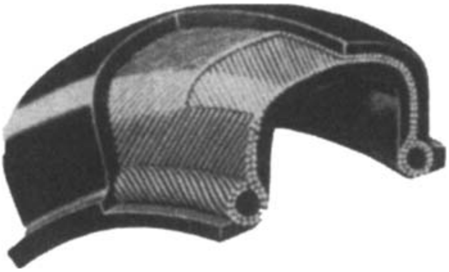


Figure 7.25 Cutaway of a rubber tire coupling. (Courtesy of Dodge Division of Reliance Electric.)

equipment. Since the torsionally soft resilient connecting member carries the full load of torque and compensates for misalignment, it is the only part of the coupling that ever needs replacement. Because there are no rubbing surfaces, dirt or other abrasive material will not shorten the life of the coupling. Problems and high lubrication costs are minimized. Torsionally soft couplings of this type have been used successfully on many drives where jamming of equipment is unavoidable. In most cases the resilient connecting member absorbs the shock load by virtue of its torsional softness or ability to assume a large angular displacement. When the shock loads reach certain levels, the resilient connecting member will slip slightly between the clamping surfaces of the hub and the clamp ring. Under extreme shock-overload conditions, the connecting member may be torn apart, disconnecting the drive and driven equipment and guarding it from damage.

Normally, the resilient connecting member is the softest, least costly, and most easily replaced part of the system. The coupling element can be replaced in 10–20 min, depending on the size and location of the unit. The heart of the tire-shaped coupling is the resilient connecting member shown in Figure 7.23, usually a fabrication of vulcanized natural rubber and plies of synthetic cord. The plies of cord are crossed alternately so that regardless of the direction of rotation, half the cord plies will be working in tension to carry the load. This torsionally soft coupling readily accommodates angular misalignment to 4°, parallel misalignment to 1/8 in., or end float up to 5/16 in. It follows that size, misalignments, loads, and speed are significant in determining the service life of the coupling selected for a given application. The design can be adapted to accommodate greater misalignments by using two elements with a shaft (Figure 7.14).

2. Bonded Shear Couplings

The basic fonded shear coupling consists of elastomeric material bonded between two hubs and is used for low-frequency isolation and accommodation of moderate parallel misalignment. (See Figure 7.26.)

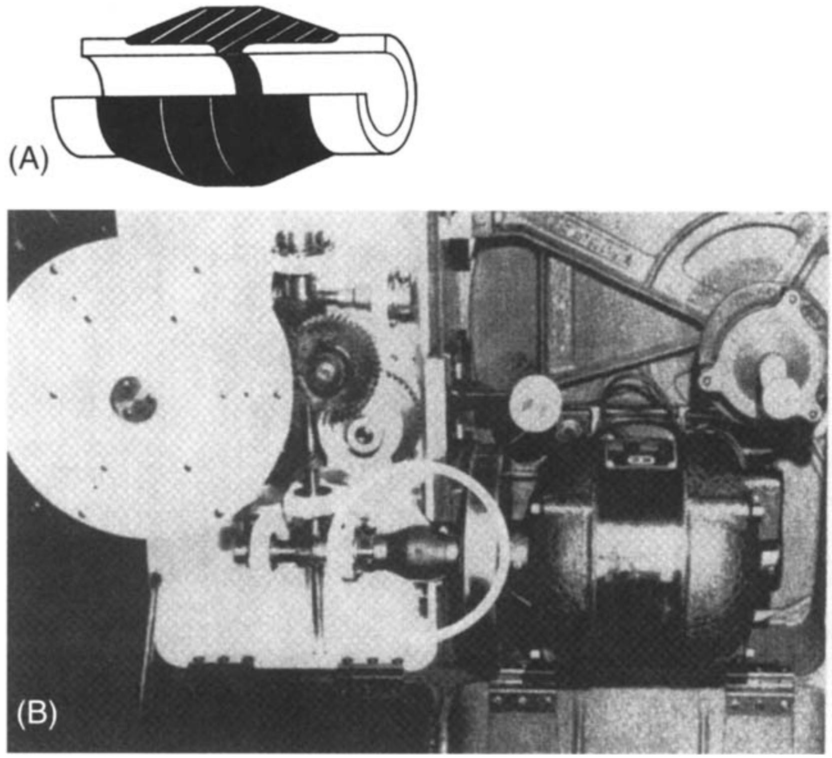


FIGURE 7.26 (A) Bonded shear coupling and (B) its applications. (Courtesy of Lord Corporation.)

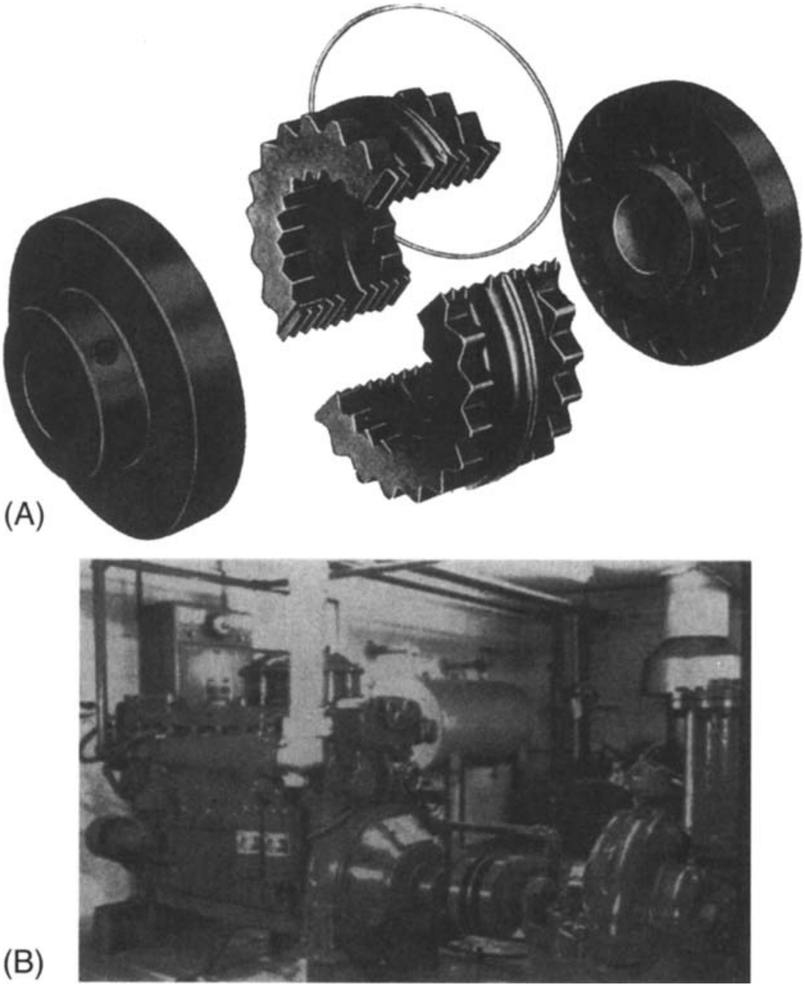
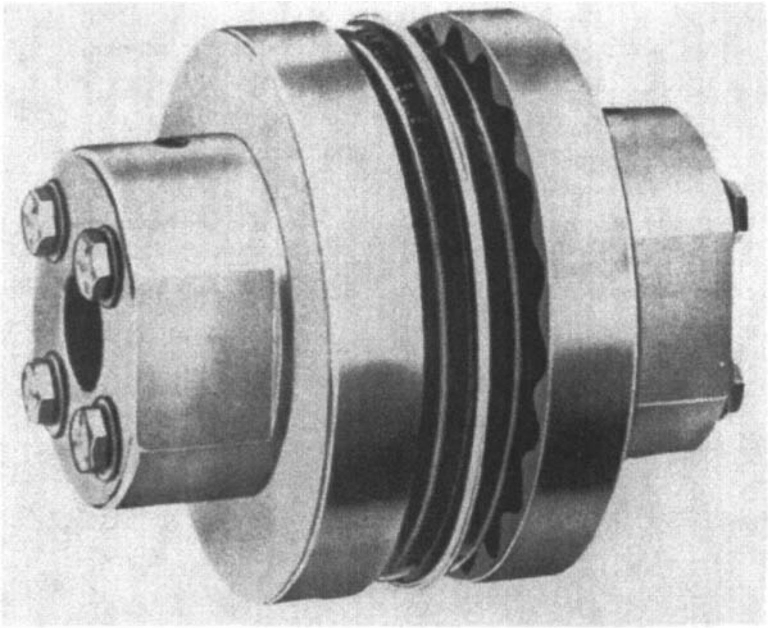


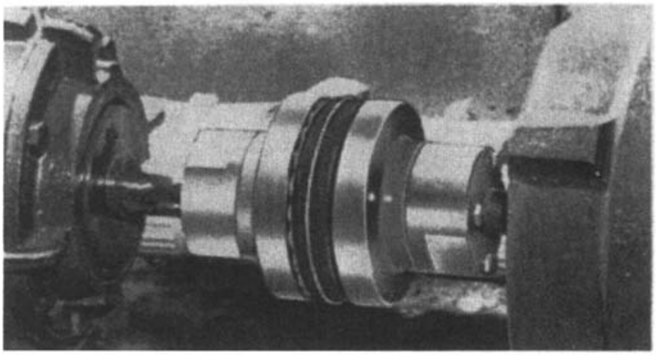
FIGURE 7.27 (A) Unclamped donut coupling and (B) application. (Courtesy of TB Wood's, Inc.)

3. Donut-Type Couplings

The elastomeric shear unclamped donut-type coupling (Figure 7.27) transmits torque by means of shear loading of the element. With a rubber element, it provides low torsional stiffness and low lateral force due to misalignments. Several materials of donut configuration can increase



(A)



(B)

FIGURE 7.28 (A) Donut spacer coupling and (B) its application. (Courtesy of TB Wood's, Inc.)

the load-carrying capacity of a given coupling size. Torsional and lateral stiffness for a given coupling size generally increase as the load-carrying capacity increases. A rubber donut permits accurate theoretical prediction of the dynamic characteristics of this coupling. This type of coupling is generally used on pumps and motors and for standard spacing requirements in the pump industry.

Donut spacer couplings allow suitable space between shafts so that pump packing can be replaced without disturbing shaft alignment. The coupling consists of two flanges, a sleeve, and two hubs (see Figure 7.28).

8

Metallic Element Couplings

I. INTRODUCTION

The development of metallic element couplings came about through a desire to eliminate the problems associated with lubricated couplings. The basic designs of most metallic element couplings have been around longer than most mechanically flexible or elastomeric couplings. Use of metallic couplings was very limited prior to the 1950s, and they did not become widely used until the late 1960s. These couplings rely on the flexure of metallic materials to accommodate for misalignment and axial movement.

There are two basic types of metallic element couplings. One is the disc coupling, which usually consists of several flexible metallic elements that are alternately attached with bolts to the opposite flanges (Figure 8.1). The other type is the diaphragm coupling, which consists of one or more metallic elements that are attached at the outside diam-

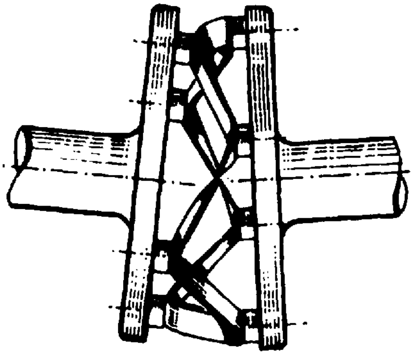


FIGURE 8.1 Disc coupling.

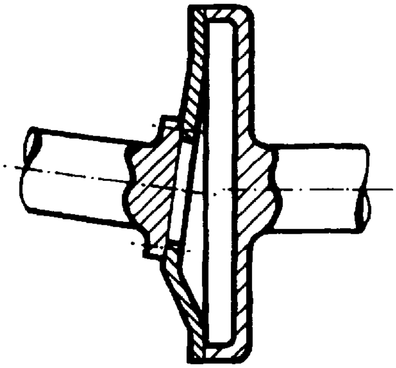


FIGURE 8.2 Diaphragm coupling.

eter of a drive flange and transfer torque through the diaphragm to an inside-diameter attachment (see Figure 8.2).

A. Stresses

Metallic element couplings are designed for infinite life. In applying flexible element couplings to industrial applications, the most important design consideration in relation to the disc, contoured diaphragm, flexible frame, and multiple convoluted diaphragm is the operating stresses in the flexible element. These stresses must be designed to be under the endurance limit of the material used.

The types of stresses that must be considered are discussed below. A multiple convoluted diaphragm is used as an example. Most flexible element couplings have similar stresses, and the approach is very similar.

1. Diaphragm Stresses

To understand diaphragm stresses, it is important to understand how the diaphragm reacts to the various types of misalignment. Figure 8.3 is exaggerated to demonstrate more clearly the diaphragm's reaction to various forces. It is also important to realize that a diaphragm on the centerline of the diaphragm pack reacts differently than a diaphragm off the centerline, when angular misalignment is imposed on the entire pack.

Some of the stresses resulting from the diaphragm deflection shown are continuous during the entire period of operation, and these are termed steady-state stresses. On the other hand, some of the stresses not only vary but go through complete reversals during each revolution. These are termed alternating stresses.

a. Steady-State Stresses. Steady-state stresses are considered to be the stresses that result from axial displacement of the diaphragm (Figure 8.3A), torque transmission, centrifugal effects, and thermal gradient effects.

Axial stress S_A is determined by the amount of deflection imposed on the diaphragm. The stiffness of a pack is equal to $K_a = W/d$, where W is axial force, and d is distance moved. The stiffness of a coupling is equal to $K_A = W/2d$.

Shear stress τ occurs when torque is transmitted through the diaphragm pack and is dependent on the size, number, and thickness of the diaphragms. Shear stress τ is highest at the inside diameter of the diaphragm. *Note:* In cases where torque is cyclic or reversing, the stress or part of it must be combined with other dynamic stresses.

Centrifugal stress S_C always results when the coupling is rotated,

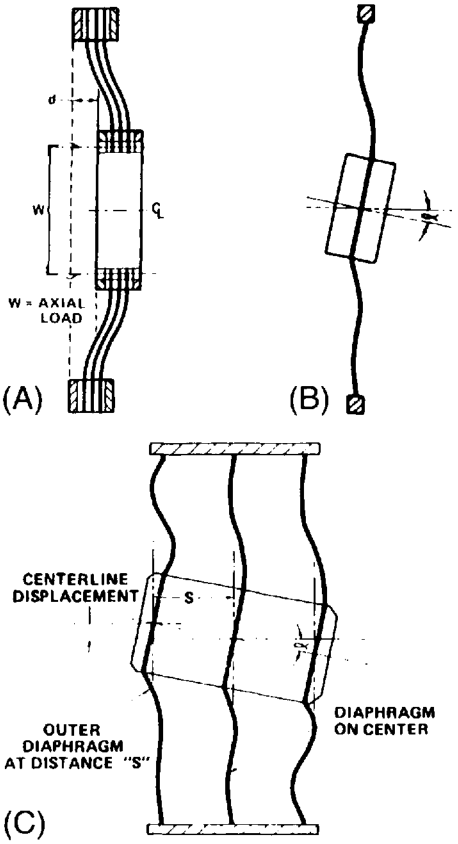


FIGURE 8.3 Reactions to stress of a diaphragm. (A) Axial displacement of a diaphragm coupling. (B) Flexure of a diaphragm coupling due to misalignment. (C) Offset displacement of a multiple diaphragm.

and this rotational effect on the diaphragm must be combined with the other steady-state stresses.

Thermal gradient stress S_t applies only where there is a temperature differential across the surface of the diaphragms and/or where there is a coefficient of expansion difference. In these cases the combined thermal stress must be calculated.

b. Alternating Stress S_B. Two cases are considered that contribute to the total alternating stress: angular flexure stress (Figure 8.3B) and stresses due to offset deflection (Figure 8.3C).

Flexure stress (Figure 8.3B) S_F is the result of the angular misalignment of the coupling. α is the misalignment angle in degrees.

Offset stress (Figure 8.3C) S_o is caused by angular deflection of the inside diameter of the diaphragm with respect to the outside diameter. The diaphragm elements that are spaced axially away from the centerline of flexure experience a stress proportional to the distance that they are removed from the centerline. Figure 8.3C shows a cross section of an exaggerated pack and how the outer diaphragms are compressed or stretched due to the distance(s) removed from the center of misalignment.

c. Combined Diaphragm Stress. All the stresses discussed above are calculated at the inside of the diaphragm and for the diaphragm farthest from the centerline, then combined in the following manner to give the highest stress point in the diaphragm pack.

Total steady-state normal stresses (S) are computed by summing the axial stress, thermal stress, and centrifugal stress:

$$S = S_A + S_t + S_c \tag{8.1}$$

This total stress is then combined with the shear stress to produce the combined steady-state stress S_M:

$$S_M = \frac{S}{2} + \sqrt{\left(\frac{S}{2}\right)^2 + \tau^2} \tag{8.2}$$

Total alternating stress (S_B) is conservatively determined by the simple summation of the offset and flexure stresses. Where no cyclic torque is present,

$$S_B = S_o + S_F \tag{8.3}$$

If cyclic torque is present,

$$S_B = \frac{S_o + S_F}{2} + \sqrt{\left(\frac{S_o + S_F}{2}\right)^2 + \tau^2} \tag{8.4}$$

Finally, the mean stress and the alternating stress resulting from bending of the diaphragm can be plotted on a modified Goodman diagram for various misalignments and total coupling axial displacements. Using a typical Goodman equation, the value of the design factor can be calculated. The results are shown in Figure 8.4.

$$\frac{1}{\text{S.F.}} = \frac{S_M}{S_{\text{ult}}} + \frac{S_B}{S_{\text{end}}} \tag{8.5}$$

where

S.F. = safety factor
S_{ult} = ultimate strength of material (psi)
S_{end} = endurance limit of material (psi)

B. System Design Considerations

When a metallic element coupling is used to connect rotating equipment, three system questions arise:

1. How can we predict the coupling's axial natural frequency?

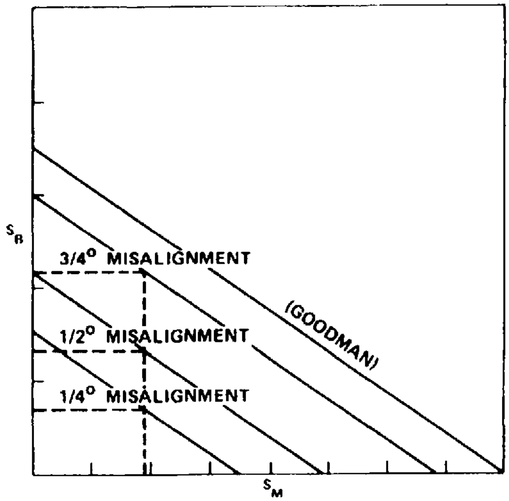


FIGURE 8.4 Modified Goodman diagram.

2. How can we analyze the coupling when calculating the lateral critical speeds of the system?
3. How much heat will be generated, and what will the operating temperature be in the coupling's enclosed guard?

1. Axial Natural Frequency

The axial natural frequency (A.N.F.) of a metallic element coupling is determined by the axial stiffness properties of the metallic element and by whether it exhibits a damped or undamped response. Single-element convoluted styles exhibit an undamped response and will have an A.N.F. value that must be considered. Multiple disc, some multiple diaphragm, and nonconvoluted single-element flexible elements exhibit a damped response, and their A.N.F. does not usually have to be considered.

The axial natural frequency is a function of the coupling stiffness and the coupling's center weight. Figure 8.5 shows two cases that could represent motion and vibration response of a diaphragm coupling as connected to a system. A response plot for case I is given in Figure 8.6. For this plot, C is the viscous damping, w the driving frequency, ω_n the

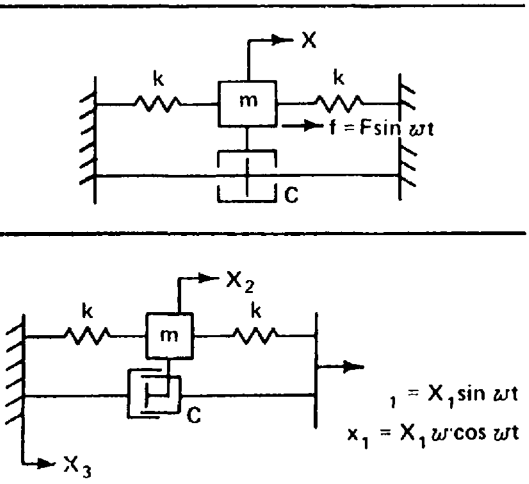


Figure 8.5 Free-body diagram of a metallic element coupling.

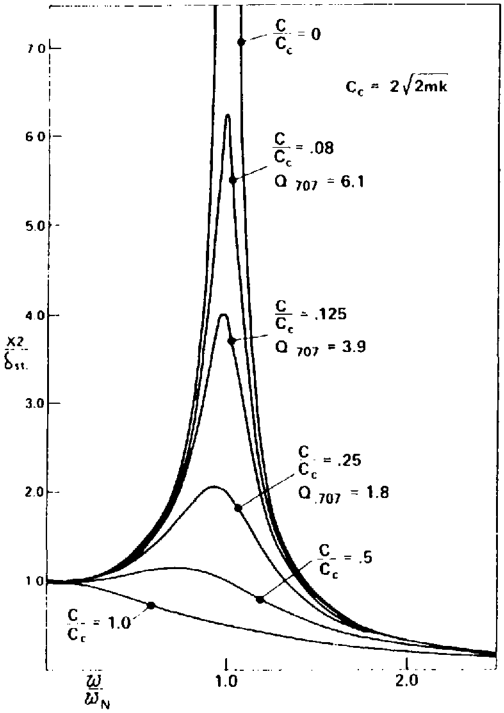


Figure 8.6 Plot of response versus forcing function.

natural frequency, F_n the natural frequency in cycles per minute, and W the weight of the center body (lb):

$$F_n = \frac{60}{} \sqrt{\frac{2K_a g}{W}} = 375 \sqrt{\frac{K_A}{W}} \tag{8.6}$$

The static deflection δ_{st} equals the driving force divided by $2K$. K_a is the stiffness of a diaphragm pack (lb-in.), and K_A is the stiffness of a coupling.

The plot in Figure 8.6 is the response to the forcing function, and the peak of the curve is defined as the magnification ratio Q. The Q for case 2 is one-half of Q for case 1. If Q for a coupling is found to be 24, and if the equipment shown is moved at a frequency equal to the natural frequency at an amplitude of 2 mils peak to peak, then the center body will vibrate at 24 mils peak to peak.

Many metallic element couplings have linear axial stiffness or exhibit linear stiffness' within certain axial travels. The following couplings exhibit linear stiffness:

1. Single convoluted diaphragm couplings
2. Some multiple convoluted diaphragm couplings

The following couplings exhibit nonlinear stiffness:

1. Tapered contoured couplings
2. Some multiple convoluted diaphragm couplings
3. Multiple straight diaphragm couplings
4. Most disc couplings

Figure 8.7 gives typical linear and nonlinear stiffness values for various couplings. For couplings that have linear stiffness, only one A.N.F. value exists. For nonlinear stiffness many A.N.F. values exist.

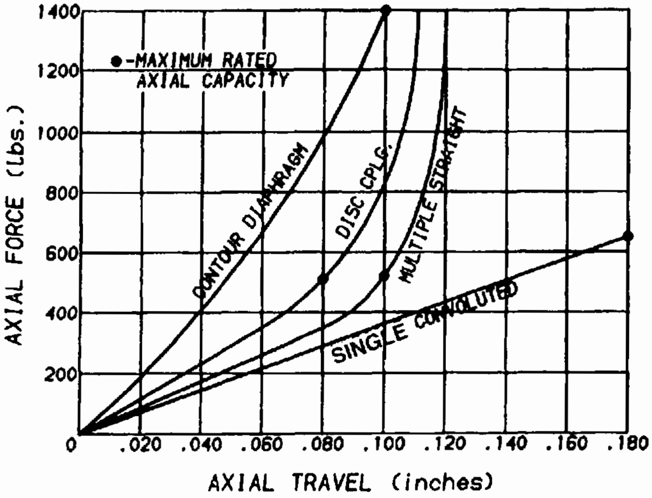


FIGURE 8.7 Linear and nonlinear axial stiffness.

Because of this, the rate of stiffness changes, and the magnitude of excitation, couplings that exhibit nonlinear stiffness rarely become excited. Therefore A.N.F. is a non-issue for this type of coupling.

2. Lateral Critical Speed

Figure 3.4A is a schematic of a coupling as connected to a system. Figure 3.4B is a free-body diagram depicting the mass–spring system that models the coupling as connected to a system.

The critical speed of a flexible element coupling is conservatively calculated by modeling the coupling as a distributed mass with pinned supports, with the supports having a stiffness equal to the lateral stiffness of the diaphragm packs.

$$y_1 = \frac{5WL^3}{384EI} \tag{8.7}$$

$$I = \frac{\pi}{64E}\,(D_o^4 - D_i^4) \tag{8.8}$$

$$N_1 = 211.4\sqrt{\frac{1}{y_1}} \tag{8.9}$$

$$y_2 = \frac{W}{K_{LL} \times N \times 2} \tag{8.10}$$

$$N_2 = 187.7\sqrt{\frac{1}{y_2}} \tag{8.11}$$

$$N_c = +\frac{1}{\sqrt{\dfrac{1}{N_1^2} + \dfrac{1}{N_2^2}}} \tag{8.12}$$

where

y_1 = tube deflection (in.)
W = weight of floating member (lb)

L = length between element centerlines (in.)
$E = 30 \times 10^6$ (psi)
I = area moment of inertia (in.^{4})
D_o = spacer tube outside diameter (in.)
D_i = spacer tube inside diameter (in.)
N_1 = critical speed of tube (cpm)
y_2 = deflection of diaphragms (in.)
K_{LL} = lateral stiffness per diaphragm (lb/in.)
N = number of diaphragms per pack
N_2 = critical speed of diaphragms (cpm)
N_c = critical speed of coupling (cpm)

The critical speed of a coupling as connected to a system must also consider the stiffness of the equipment shaft and the equipment bearing stiffness. These stiffnesses tend to lower the actual measured critical speed and can be calculated as follows:

$$N_S = \frac{30}{\pi}\sqrt{\frac{K_e}{W_c}} = \text{system critical speed (cpm)} \tag{8.13}$$

$$\frac{1}{K_e} = \frac{1}{2(K_c/2)} + \frac{1}{2K_{LL}} + \frac{1}{2K_S} + \frac{1}{2K_B} \tag{8.14}$$

$$N_C = 211.4\sqrt{\frac{1}{\Delta}} \tag{8.15}$$

$$\Delta = \frac{5WL^3}{384EI} \tag{8.16}$$

$$K_c = \frac{W_c}{g}\left(\frac{\pi N_c}{30}\right)^2 \tag{8.17}$$

where

K_S = equipment shaft stiffness (lb/in.)
K_B = equipment bearing stiffness (lb/in.)
W_c = coupling weight (lb)
K_c = stiffness of coupling shaft (lb/in.)

3. Heat Generation and Windage Loss

All flexible shaft couplings must be surrounded by a guard in order to comply with OSHA requirements. For most high-speed applications, these guards are oiltight enclosures and their purpose is to trap any oil that escapes from the machinery seals. When couplings rotate at high speeds in these oiltight enclosures, they shear the air, and this shearing results in significant heat generation. The temperature of the enclosure, as well as that of the coupling, depends on the amount of air shearing and on the ability of the enclosure to dissipate the heat. Another problem with couplings operating in oiltight enclosures is that negative pressures are occasionally created in the vicinity of machinery seals, causing oil to be sucked into the enclosure, which is particularly detrimental for gear reducers.

Numerous tests have been performed by many coupling manufactures to calculate the enclosure temperature. There are several empirical methods that can be used. Some are more accurate than others, but they usually give a conservative estimate of the maximum temperature that may be seen by the enclosure.

a. When Are Coupling Enclosures a Problem? Coupling enclosures were not much of a problem with oil-lubricated couplings. The design was relatively simple and was usually assigned to a junior engineer. But with the application (particularly retrofits) of flexible element type couplings in these oiltight enclosures, hot enclosure problems became more prevalent. For a short time, until more thought was put into the design, oil lines were reconnected and some dry couplings were cooled with oil spray.

b. How Is Heat Generated? Heat in a coupling enclosure is generated by the shearing of the air (Figure 8.8). The air in an enclosure is contained in an annular cylinder with the outer diameter (O.D.) equal to the enclosure diameter, the inner diameter (I.D.) equal to the coupling's diameter, and a length equal to that of the enclosure. Heat is generated by air shearing and/or turbulence. There are two types of frictional losses: disk loss and cylinder loss.

c. Temperature Prediction. Several methods can be used to predict enclosure temperatures. Two are presented here. These methods gener-

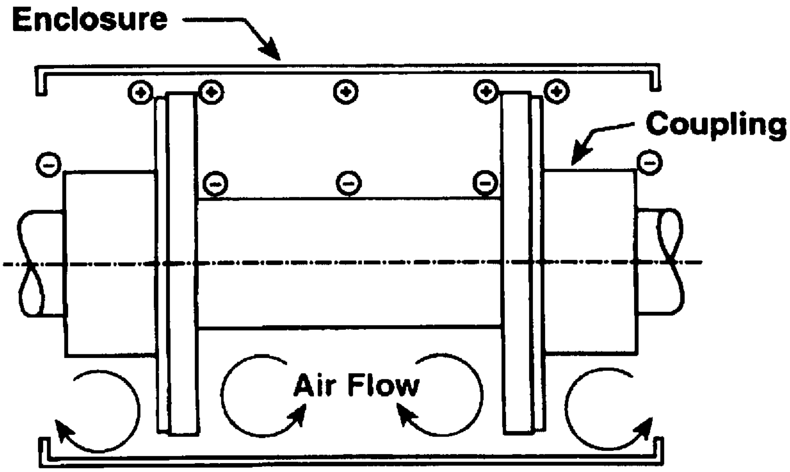


FIGURE 8.8 Disk effect inside an enclosure. Note the distribution of positive and negative pressure due to the disk effect.

ally yield values within ±15% of actual measured values (see Figure 8.9).

HORSEPOWER LOSS METHOD

The horsepower (hp) loss method calculates the temperature rise of a coupling guard due to rotation of the coupling losing energy to the surrounding air in the form of heat. The algorithm used models the coupling as disks and cylindrical sections with specific diameters and lengths (see Figure 8.10). *Note:* The temperature of the air inside the guard may tend to be higher than the guard temperature due to frictional heating, but the guard temperature is assumed to be the air temperature to be conservative. Two types of losses are considered in this method: disk windage loss and cylinder windage loss.

1. Disk windage loss. The disk windage power accounts for frictional losses in both ends of the guard. The correlation with rpm and diameter is

$$\text{Disk windage power} \approx \text{rpm}^3 \times \text{diameter}^5$$

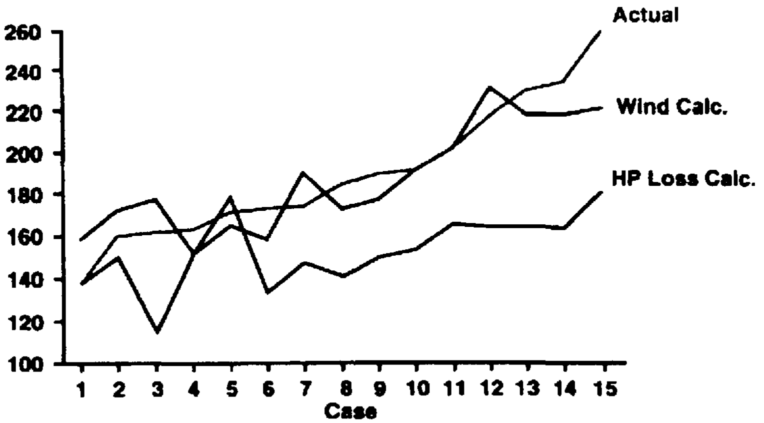


Figure 8.9 Temperature comparison graph.

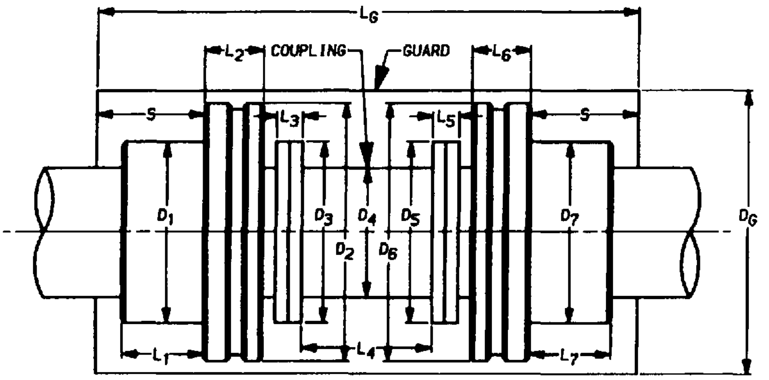


Figure 8.10 Coupling outlined in a coupling guard.

The equation to find disk windage loss for a coupling is

$$hp_{\text{loss disk}} = \text{rpm}^{2.85}\frac{1}{K_1}\left(\frac{S}{D_W}\right)^{1/10} \tag{8.18}$$

where D_W and K_1 are found on the horsepower loss constant chart (Figure 8.11) for the proper size of coupling, S (in.). Equation (8.18) is to be applied to each end of the coupling if disk loss is present. If S/D_W is greater than 1.0, use $S/D_W = 1.0$.

D (in.)	$K_1 \times 10^{10}$
6	2460
9	372
12	67.6
16	19.3
24	4.76

FIGURE 8.11 Typical horsepower loss constant chart.

2. *Cylinder windage loss.* The cylinder windage power correlation with rpm, diameter, and length is

$$C_{\text{losses}} = \text{rpm}^3 \times \sum \text{diameter}^4 \times \text{length}$$

The equation to find the amount of horsepower loss for each cylindrical section is

$$\text{hp}_{\text{loss cylinder}} = EL_C C_f \tag{8.19}$$

where

$E = \text{rpm}^3 \times D_C^{3.859} \times 5.5 \times 10^{-15}$
D_C = diameter of section (in.)
L_C = length of section (in.)
C_f = cylinder friction coefficient

$$C_f = \frac{128B + 2.075}{B \times \text{rpm} \times D_C^2} + 0.0015 \tag{8.20}$$

where

$$B = \frac{D_G - D_C}{D_C} \tag{8.21}$$

D_G = diameter of guard section (in.)

These calculations are done for each section, and all the $\text{hp}_{\text{loss cylinder}}$ values calculated are summed together.

3. *Total windage loss.* Add the disk hp loss and the sum of cylinder hp loss loss to find the total hp loss.

$$\text{hp}_{\text{loss total}} = \text{hp}_{\text{loss disk}} + \sum \text{hp}_{\text{loss cylinder}}$$

To find the assumed temperature of the guard, take the area of the guard, A_g (ft^2), and divide it into the hp loss total,

$$A_g = \frac{\pi}{144} \times D_G \left(L_G + \frac{D_G}{2} \right) \quad (\text{ft}^2) \tag{8.22}$$

where

D_G = diameter of guard (ft)
L = length of guard (ft)

From Figure 8.12 for the total hp/A_g and the correct ambient temperature, T_a (°F), find T_{gi}, the assumed temperature of the guard (°F). If $T_{gi} > 175$°F, use this scaling technique to find the actual horsepower loss and operating temperature of the guard

$$\text{hp}_{\text{loss total}} = \frac{590}{T_{gi} + 560} = \text{hp}_{\text{loss actual}} \tag{8.23}$$

Again from Figure 8.12, use $\text{hp}_{\text{loss actual}}/A_g$ and ambient temperature to find the operating temperature of the guard (°F).

WINDAGE METHOD

This method is mainly empirical, having been derived from R&D testing.

Coupling temperature with air cooling: All dimensions are in inches, and the temperatures are in degrees Fahrenheit (see Figure 8.13).

For enclosures with air cooling, the coupling temperature is given by the equation

$$T_c = K_2 K_3 K_4 \left(\frac{S}{1000} \right)^{K_{4e}/K_1} + \frac{T_A + T_S}{2} \tag{8.24}$$

where

T_c = coupling temperature
K_1 = air shear factor = [(enclosure mean dia.)/(max. coupling dia.)]$^{0.27}$
K_2 = area factor = (coupling surface area factor)/(enclosure surface area)

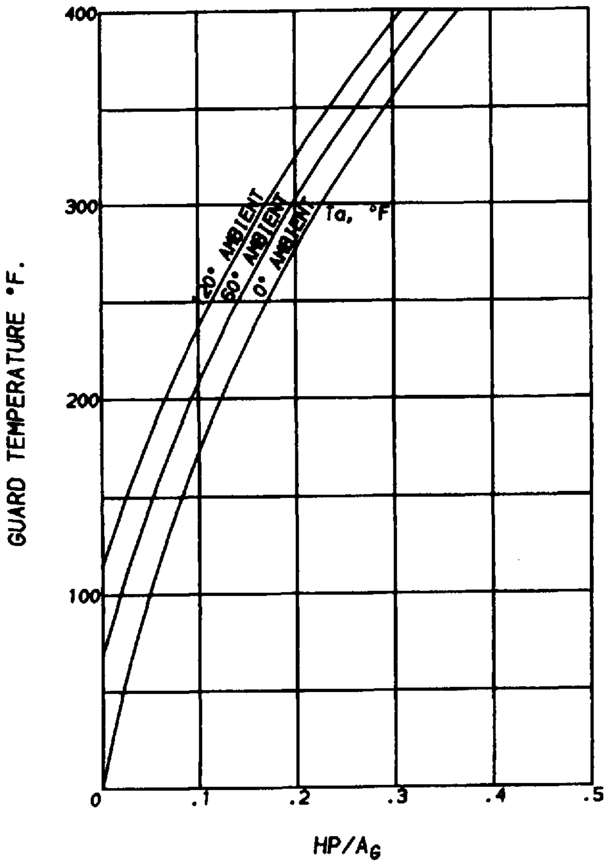


Figure 8.12 Temperature rise versus hp/A_g.

K_3 = coupling circulation factor = [(max. coupling dia.)/(min. coupling dia.)]$^{0.2}$

K_4 = (1/1000) × (enclosure surface area/exhaust port area)

S = application speed (rpm)

T_A = ambient temperature

T_S = shaft temperature

and the exponential factor K_{4_e} is given by the empirical formula

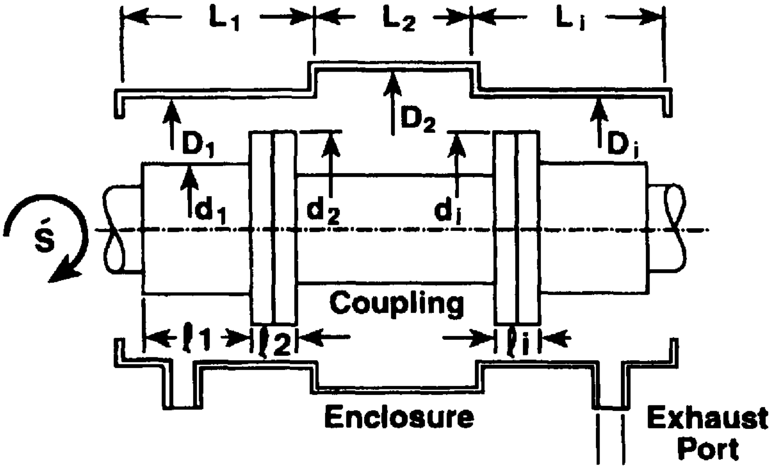


FIGURE 8.13 Inputs for windage method.

$$K_{4e} = \left(\frac{1}{K_4}\right)^{0.27} + 0.65 \tag{8.25}$$

Observations: The coupling surface area (K_{ac}) used to determine K_2 can be evaluated with the formula

$$K_{ac} = \sum_{i}^{N_c}=1\, d_i^{2.8} l_i \tag{8.26}$$

where

K_{ac} = coupling surface area factor
N_c = number of coupling cylinder surfaces
l_i = length of cylinder i
d_i = diameter of cylinder i

The enclosure surface area is

$$A_e = \pi \sum_{i=1}^{N_e} L_i D_i \quad (\text{in.}^2) \tag{8.27}$$

where

A_e = enclosure surface area
N_e = number of enclosure cylindrical surfaces
L_i = length of cylinder i
D_i = diameter of cylinder i

The enclosure diameter to be used in equation (8.24) is given by

$$D_{\text{em}} = \frac{\sum_{i=1}^{N_e} L_i}{\sum_{i=1}^{N_e} \frac{L_i}{D_i}} \tag{8.28}$$

where D_{em} = enclosure mean diameter.

The maximum value for K_4 is 0.6 (case of no air flow), while a properly designed enclosure will exhibit an air flow factor $K_4 = 0.1–0.2$. Mathematically, the exponential factor K_{4_e} must be situated in the range of 1.8 (for $K_4 < 0.2$) to 2.2 (for $K_4 = 0.6$).

The exhaust ports area is given by the equation

$$A_{cp} = \frac{\pi}{4} \sum_{i=1}^{N_p} D_{pi}^{\;2} \tag{8.29}$$

where

A_{cp} = exhaust ports area
N_p = number of exhaust ports
D_{pi} = diameter of exhaust port i

d. Guard Design Considerations. The following items must be considered when designing an enclosure with air cooling:

1. Sizing the air ports
2. Separating the oil from the air
3. Avoiding negative pressures

1. *Sizing the air ports.* The exhaust port should be tangential to the enclosure's outside diameter and directed with the coupling rotation

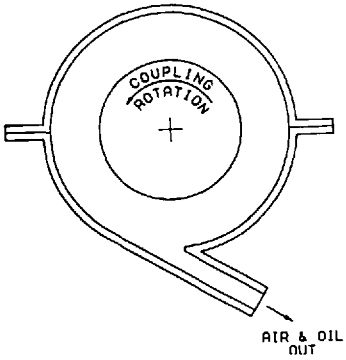


FIGURE 8.14 Exhaust port.

(see Figure 8.14). This port (or ports) can have any profile; however, a tubular port is the most convenient to use. The larger the port, in comparison with the enclosure, the lower the enclosure's temperature. A large exhaust port, on the other hand, can create problems in separating the oil from the air. The intake and exhaust ports should be placed so that no "dead zone" is left in the enclosure. Hence, if one exhaust port is placed in the middle of the enclosure's length, there should be one intake port at each end of the enclosure. Conversely, if two exhaust ports are used (which is desirable for enclosures longer than 36 in.), then one intake port should be placed halfway between the exhaust ports. The total intake port area should be half the total exhaust area. For example, if one exhaust port of 3 in. diameter is used, then each of the two intake ports should be 1.5 in. in diameter.

2. *Separating the oil from the air.* The amount of oil that escapes past a seal is usually very small. Some equipment has no seals and the bearing oil must be recaptured from the coupling enclosure rather than from the equipment housing. It is therefore possible that oil is still flowing into the enclosure at flow rates of up to 3 gpm, even for flexible element couplings. This is hot oil and does not provide cooling.

To prevent this oil from becoming a mist, many enclosures rely on the centrifugal effect created by the rotation of the air, similar to the working of a cyclone separator. One way to increase the air rotation is to install the intake ports tangentially to the enclosure, as shown in Fig-

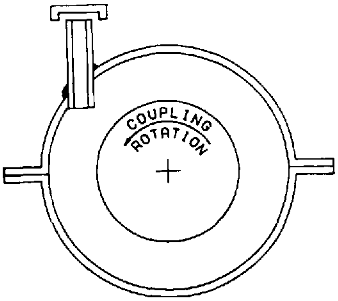


FIGURE 8.15 Tangential intake port.

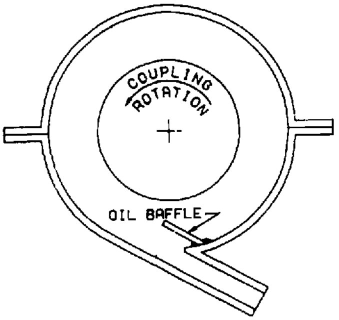


FIGURE 8.16 Oil baffle at exhaust port

ure 8.15. Another way to prevent the formation of oil mist is to shorten the residency time of any oil particle in the enclosure. An oiltight baffle should be installed along the enclosure, over the exit port, as shown in Figure 8.16. This baffle will force the oil out of the enclosure and prevent it from rotating more than one revolution before exiting.

The actual separation between air and oil is done outside the enclosure, as shown in Figure 8.17. All ports should have breather caps provided with a coarse wire mesh filter.

3. *Avoiding negative pressure.* When large-diameter disks are mounted on shafts in the vicinity of the seals, their rotation generates a negative pressure that causes oil to be sucked out from the machine housing into the coupling enclosure. This phenomenon is particularly detrimental to gear boxes, which have an independent oil supply. As

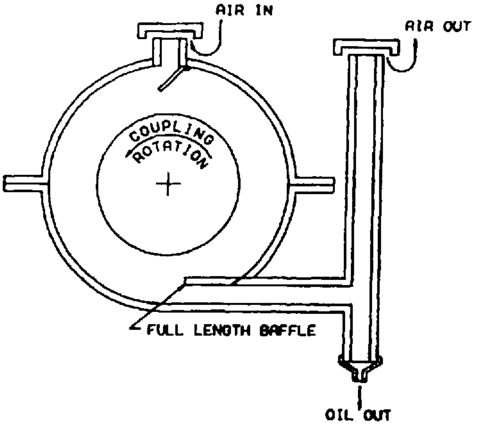


Figure 8.17 Separation of oil from air.

couplings are mounted closer and closer to the bearings (in order to reduce the overhung moment), the problem of oil misting into the enclosure is encountered more often.

To avoid this problem, an air tube can be installed close to the coupling, as shown in Figure 8.18. This tube allows atmospheric pressure near the seal and cancels negative pressure. Preferably, this air tube should be used in addition to the normal air inlet port rather than instead of it. If this tube does not eliminate the oil loss, then it should be connected to a positive pressure area of the enclosure, usually over the largest rotating part, as indicated in Figure 8.19.

e. Basic Enclosure Design Features. Usually coupling enclosures are cylindrical and split on a diametral plane. The two half-cylinders are connected by bolted flanges. A few basic design rules should be followed.

1. A coupling enclosure should be free to float at both ends. Some designs are bolted to one machine and are free to move axially at the other end. This allows for axial motion between machines, but the enclosure is rigid with respect to any offset motions.
2. A coupling enclosure should be oiltight. Even though enclosures are usually of welded construction, oil can still leak out,

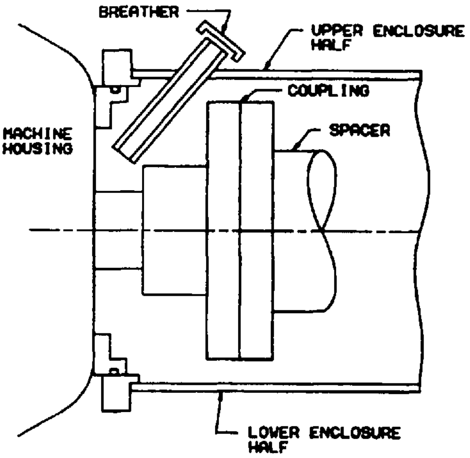


FIGURE 8.18 Compensation of negative pressures.

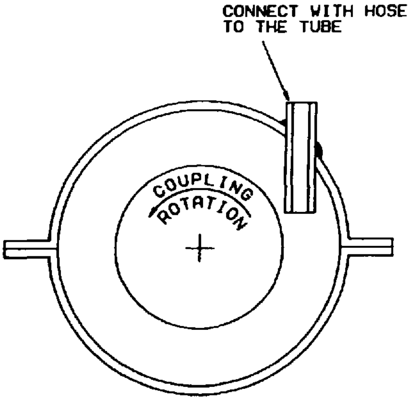


FIGURE 8.19 Means of pressurizing the seal area.

usually at the flanges. Gaskets and caulking seldom can stop the leaks. The simple design shown in Figure 8.20 can help eliminate these leaks.

3. Breathers are required on enclosures to prevent the formation of an "oil plug" and prevent the possibility of filling the enclosure with oil. Breathers should be designed so that oil cannot escape. In designing the enclosure, the direction of rotation of

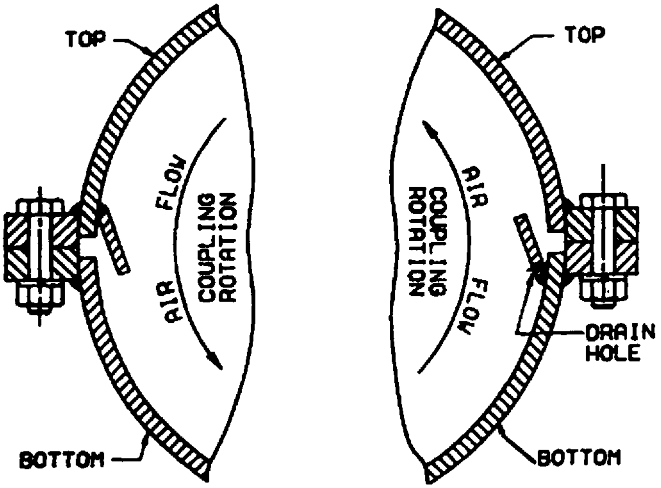


FIGURE 8.20 Seals for enclosure flanges.

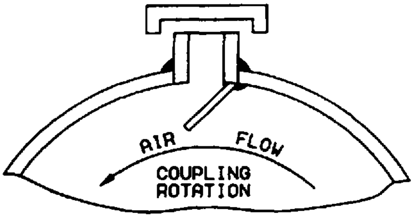


FIGURE 8.21 Baffle of intake port.

the coupling should be considered. A baffle should prevent the air from flowing out from the breather, as shown in Figure 8.21.

II. DISC COUPLINGS

The flexible disc coupling is available in a number of forms, but they all have one thing in common: The driving and driven bolts are on the same bolt circle. The types shown in Figure 8.22 are some of the most commonly used disc shapes. The flexibility that each type can attain is dependent on the free span of the material between the bolts. Torque is transmitted by the pull between the driving and driven bolts.

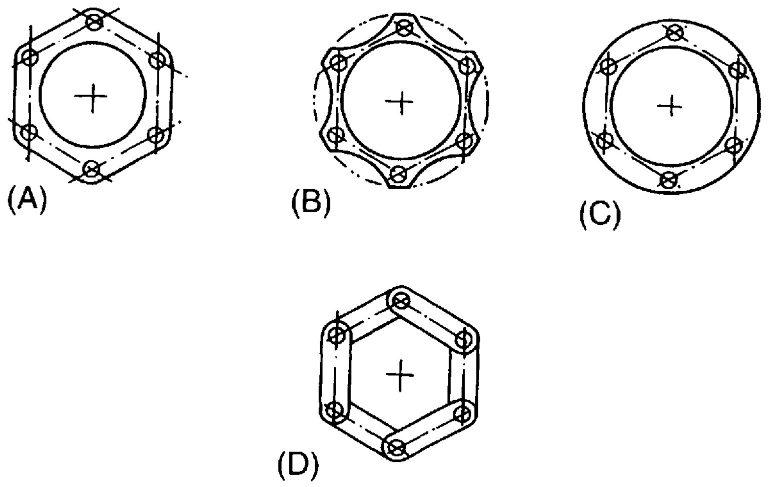


FIGURE 8.22 Various disc shapes.

The loading of the disc in Figures 8.22A and 8.22B will produce a simple tensile stress in the blades. The disc in Figure 8.22C will produce compression stresses in the outer diameter and tension stress at the inner diameter. If the line of action between the driving and driven bolts falls outside the material of the lamination, the tensile stress could be high. To ensure that this line of action falls in an acceptable position, it is sometimes necessary to use more driving bolts than would otherwise be required, and this reduces flexibility. An element of such a design requires the use of more laminate material than would otherwise be required, or a larger diameter, to reduce the stress levels. The disc in Figure 8.22B has the inoperative portion of the element removed. An attempt has been made to produce a shape that gives a uniform tensile stress pattern over the whole of the driving portion of the blade.

Bending stresses in the lamination due to misalignment will occur at the anchor points, and these will be very heavily influenced by the tensile pull in the blade. Any reduction in the tensile stress that can be made in this area will therefore be of advantage and leave a greater margin for the fluctuations in bending stress due to angular misalignment. The form shown in Figure 8.22A attempts to do this by allowing more material in the laminations in the area where the largest bending stresses occur, with the result that tensile stresses will be lower in the region of the anchor points for a given stress in the midspan of the blade. The disc transmits torque by a simple tensile force between alternate driving and driven bolts on a common bolt circle diameter and flexibility that is derived from the free span between adjacent bolts and will vary as the cube of this length.

When a portion of the flexible element between bolts is considered as a beam, the advantages of a thin laminate construction, as opposed to one thick disc, will be obvious in terms of flexibility and forces transmitted due to misalignment. Certain manufacturing considerations make it impractical to use very thin laminations for discs. Typically, thicknesses ranging from 0.005 to 0.025 in. have been found to be satisfactory. The form in Figure 8.22D has some advantages in manufacturing and with larger units also aids the assembly and disassembly of the links into the coupling.

The degree of flexibility required and the limits of acceptable bearing loads determine the number of driving and driven bolts used. This can be an important factor when selecting a coupling, as by choosing a unit with a greater degree of freedom than is actually required, not only will the diameter and weight be larger than necessary, but the cost could also be too high. It is possible for a coupling of this type to have any even number of bolts, but the simplest form, with two driving and two driven bolts, will provide the greatest degree of flexibility and generate the lowest forces. At the other extreme, it is practical to have a large number of bolts, giving a short flexing length, where it is necessary to restrict the movement of a rotor with no axial location while still allowing for some small misalignment.

For high-speed applications it is common for the disc to be assembled with "prestress." The disc pack is installed with some tension in all legs. This tension is accomplished by making the bolt circle in the hubs slightly larger than the one in the disc. This tension does two things: It ensures proper piloting for the disc pack by eliminating bolt hole clearances, and it puts pretension in the compression links, which eliminates axial "tin canning." Axial tin canning in a disc pack can occur if clearances exist between the laminations and the bushing, and then under torque the laminations slip, and when the coupling is shut down the compression side of the disc will appear buckled. In this condition one can, with very little force, move the coupling back and forth. And in some cases the coupling can be pushed into two "neutral" positions (like the bottom of a tin can). This condition (let's call this *nonpermanent tin canning*) will go away if one loosens the disc pack bolts and properly retorques them. If the disc pack bolts are loosened and retorqued and the tin canning is still there, the disc lamination may have been permanently deformed (let's call this *permanent tin canning*) from being overtorqued or axially overstretched. Permanent tinning canning can cause the disc pack to fail prematurely. For applications where there are thrust bearings, nonpermanent tin canning is usually okay. Because, under torque, the tension links will position the coupling properly and because no real damage to the disc has occurred, the disc should not prematurely fail. But on applications such as sleeve bearing motors, even nonpermanent tin canning can be a problem (although not to the coupling) because detrimental rubbing of the motor rotor during startup might occur (the coupling may not properly position the rotor if it is tin-canned). In this type of application the coupling may need to be sized on the motor starting torque to prevent "good tin canning" from the start-up torques.

A. Construction of Disc Coupling

The disc coupling is used for a large size range of drives, from fractional horsepower to very large (100,000 hp). In general we can categorize disc couplings into two groups, one for general-purpose applications and one for high-speed applications.

The general-purpose disc coupling's torque-transmission components are made of low- to medium-carbon steels. Its flexing discs are usually made of spring steel, AISI 1050 to 1080, 300 Series stainless steel. The high-speed disc coupling's torque-transmission components are usually made of alloy steels. Its flexing disc is usually made of corrosion-resistant steel, AISI 300 series stainless steel, PH stainless steel, or high-strength nickel alloy.

B. Applications of Disc Couplings

Disc couplings are used in a variety of applications. The most common is on medium-horsepower pumps. They are also used on marine drives, cooling tower drives, generators, compressors, mill equipment, fans, and machine tools. See Figures 8.23–8.26 for some typical applications.

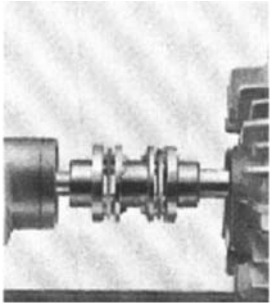


Figure 8.23 Centrifugal pump application of disc coupling. (Courtesy of Rexnord, Thomas Couplings.)

Figure 8.24 Marine diesel application of disc coupling. (Courtesy of TB Wood's, Inc.)

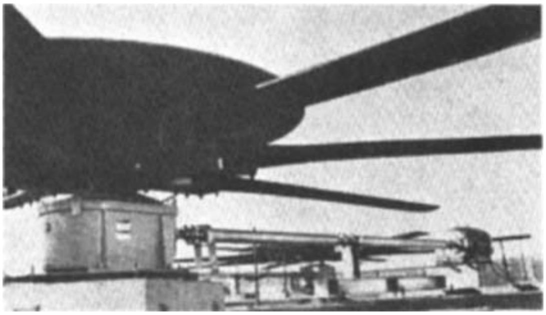


Figure 8.25 Cooling tower application of disc coupling. (Courtesy of TB Wood's, Inc.)

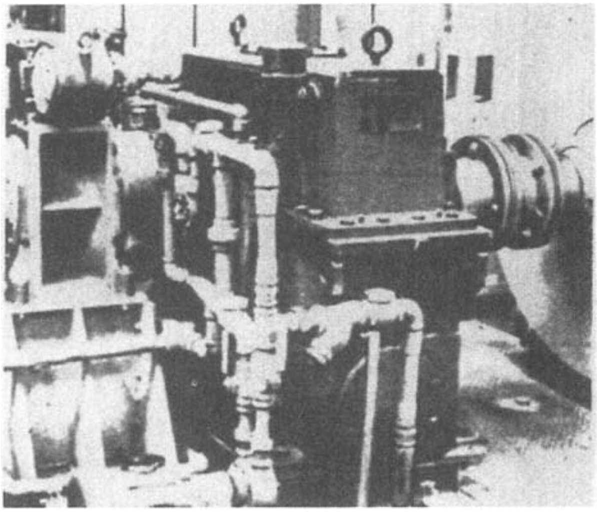


FIGURE 8.26 Generator application of disc coupling. (Courtesy of Rexnord, Thomas Couplings.)

C. Types of Disc Couplings

There are many types of disc couplings, including

A closed-couple double flex disc coupling (Figure 8.27) usually used for low-torque applications.

A floating shaft disc coupling (Figure 8.28) typically used to connect long spans.

A spacer-type disc coupling with replaceable disc packs (Figure 8.29) typically used on medium-horsepower pumps.

A spacer disc coupling using a square rather than a circular disc (Figure 8.30).

A spacer disc coupling that uses individual links rather than a one-piece disc (Figure 8.31).

A spacer disc coupling similar to the above except that the discs are square and several discs are used between the driving and driven flanges to increase flexibility (Figure 8.32).

A spacer disc coupling that uses a unitized pack to help maintain

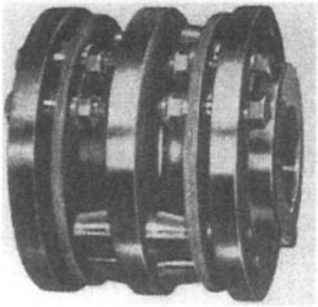


FIGURE 8.27 Closed-couple disc coupling. (Courtesy of Rexnord, Thomas Couplings.)

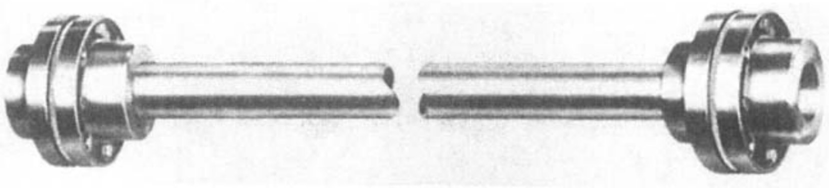


FIGURE 8.28 Floating shaft disc coupling. (Courtesy of Rexnord, Thomas Couplings.)

balance and decrease the number of parts during assembly and disassembly (Figure 8.33).

D. Failure Modes for Disc Couplings*

The most common form of failure is disc fatigue due to excessive flexure. This is usually caused by poor initial alignment of the connected machines. It can also be brought about by operational conditions.

Machines connected with flexible couplings should be aligned with the greatest possible accuracy. The better the initial alignment, the more capacity the coupling has to take care of subsequent operational misalignment. Changes from the initial condition can occur through bearing wear, settling of foundations, base distortion due to torque ther-

---

*This section was supplied through the courtesy of the Coupling Division of Rexnord, Thomas Couplings.

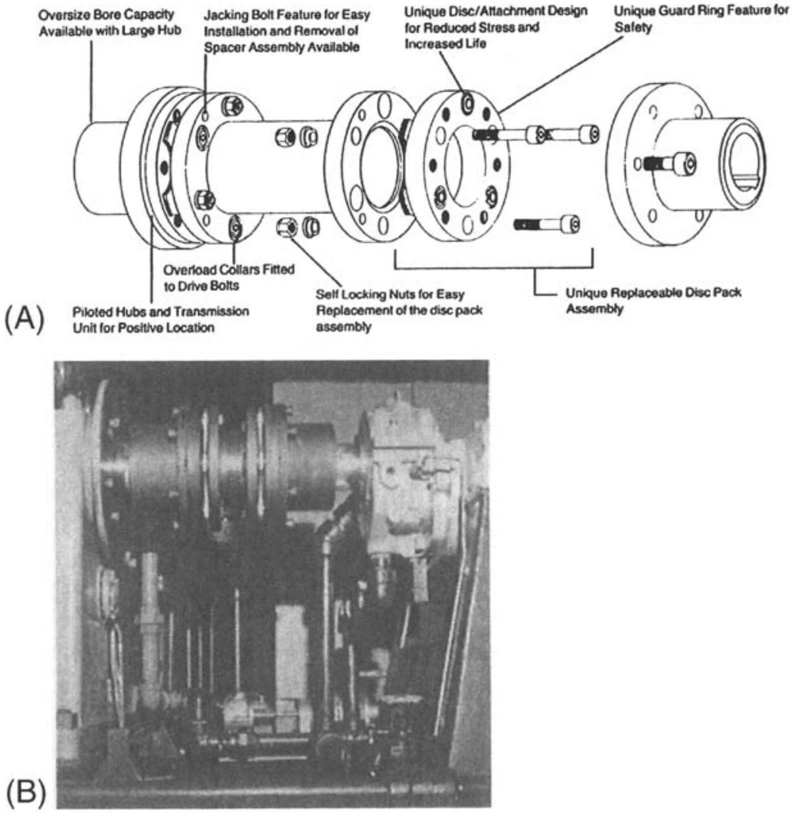

Oversize Bore Capacity Available with Large Hub
Jacking Bolt Feature for Easy Installation and Removal of Spacer Assembly Available
Unique Disc/Attachment Design for Reduced Stress and Increased Life
Unique Guard Ring Feature for Safety
Overload Collars Fitted to Drive Bolts
Piloted Hubs and Transmission Unit for Positive Location
Self Locking Nuts for Easy Replacement of the disc pack assembly
Unique Replaceable Disc Pack Assembly
(A)
(B)

FIGURE 8.29 Spacer-type disc pack coupling with replaceable disc packs. (Courtesy of Flexibox International, Metastream Coupling.)

mal changes, and vibrations in the connected machines. The alignment of the machines should be checked at regular intervals and corrected as necessary.

In Figure 8.34A the disc is broken adjacent to the washer face. This usually indicates excessive shaft misalignment during operation. This type of disc failure generally starts with the outer disc in the pack and progresses through the disc pack. The equipment must be realigned and the disc pack replaced. A hot check of alignment should be made to ensure that it is within coupling misalignment capacity.

In Figure 8.34B the disc is broken adjacent to the washer face,

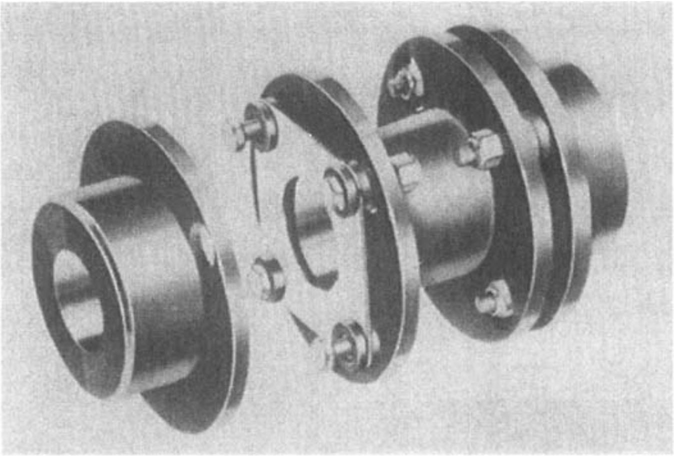


Figure 8.30 Spacer disc coupling with square disc. (Courtesy of TB Wood's Inc.)

with heavy corrosion present along the area of the break. Also, iron oxide is probably bleeding from the disc pack. This failure is typical of discs that have been in service for several years and/or have been operating in a corrosive atmosphere. Breaks first appear in the outer discs and progress into the disc pack. The disc pack should be replaced and the equipment realigned. If excessive corrosion exists on other areas of the disc or coupling component parts, they should be replaced with proper corrosion protection: stainless steel plating, painting, and so on.

The disc coupling is easily inspected. Visual analysis may point to possible drive system problems. Proper evaluation of the disc packs and connecting parts may save considerable maintenance costs and downtime. Following are some of the more evident visual inspection criteria and recommended corrective procedures.

1. The disc is broken through the bolt hole (Figure 8.34C). This indicates loose coupling bolts. Replace the disc pack and tighten bolts to the specified torque value.
2. Discs are embedded in the bolt body. This is usually the result of a loose bolt or a severe torque overload. This may also appear when the bolt is turned during installation. Replace the

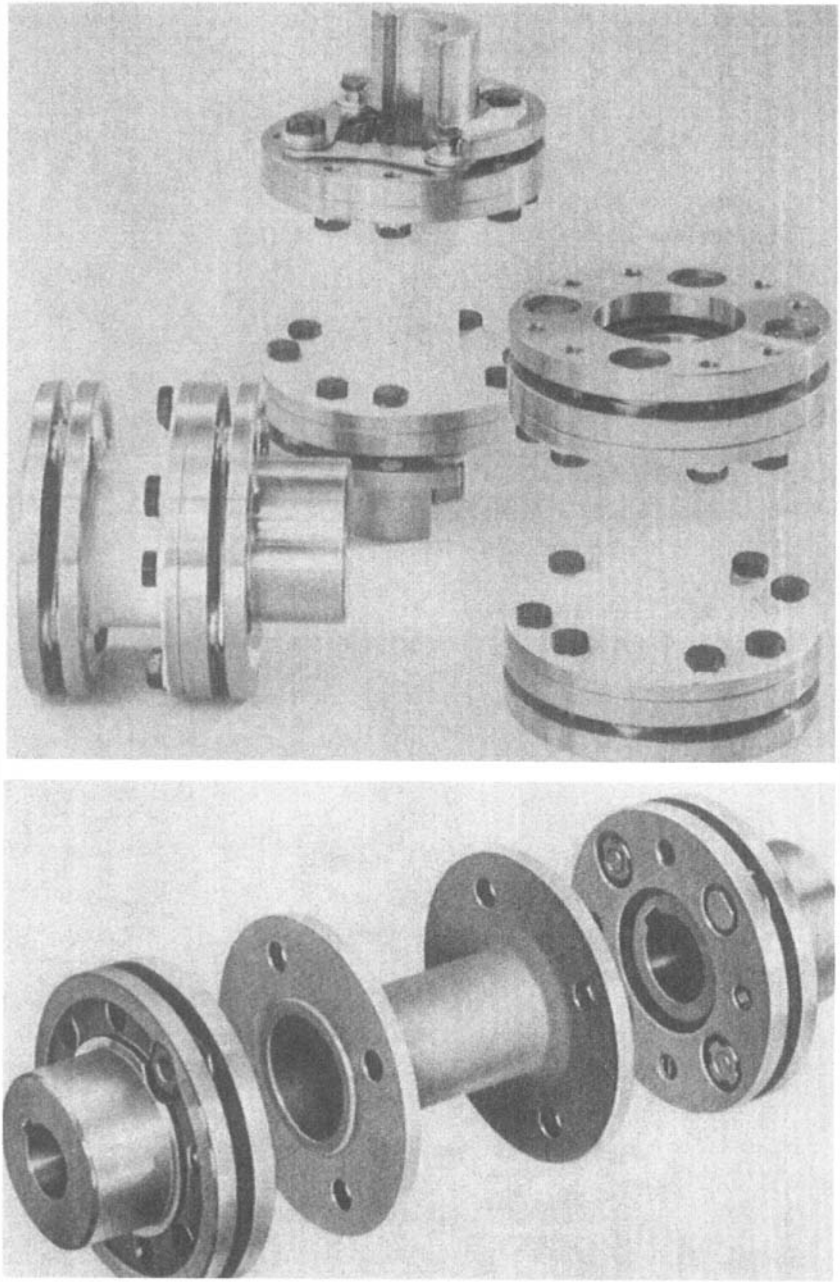


FIGURE 8.31 Spacer dic coupling with separate links. (Courtesy of TGW Thyssen Getriebe.)

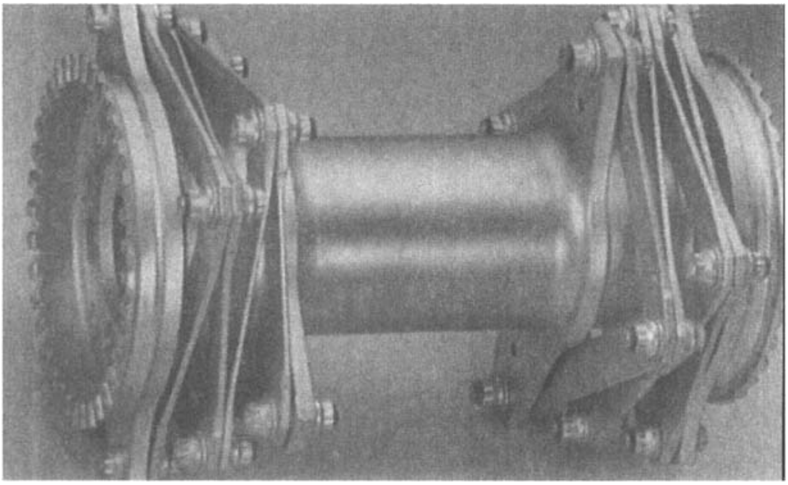


Figure 8.32 Spacer disc coupling with multiple frames. (Kaflex Coupling, courtesy of Kamatics Corporation.)

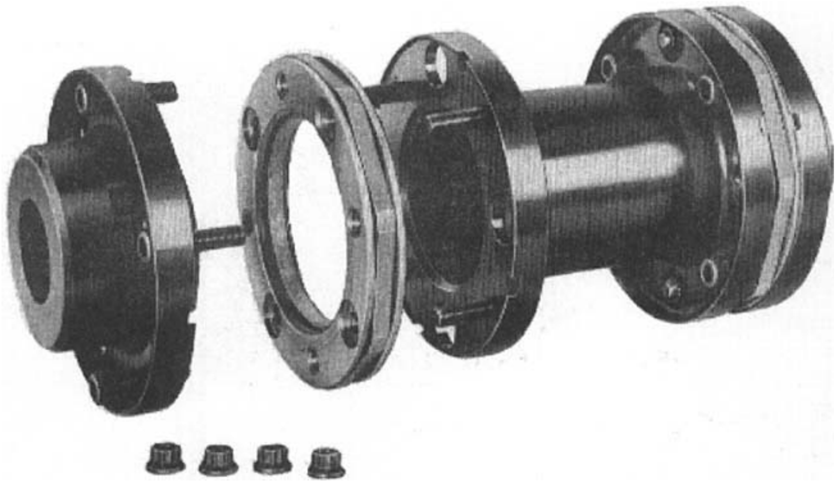


Figure 8.33 Spacer disc coupling with unitized packs. (Courtesy of Rexnord, Thomas Couplings.)

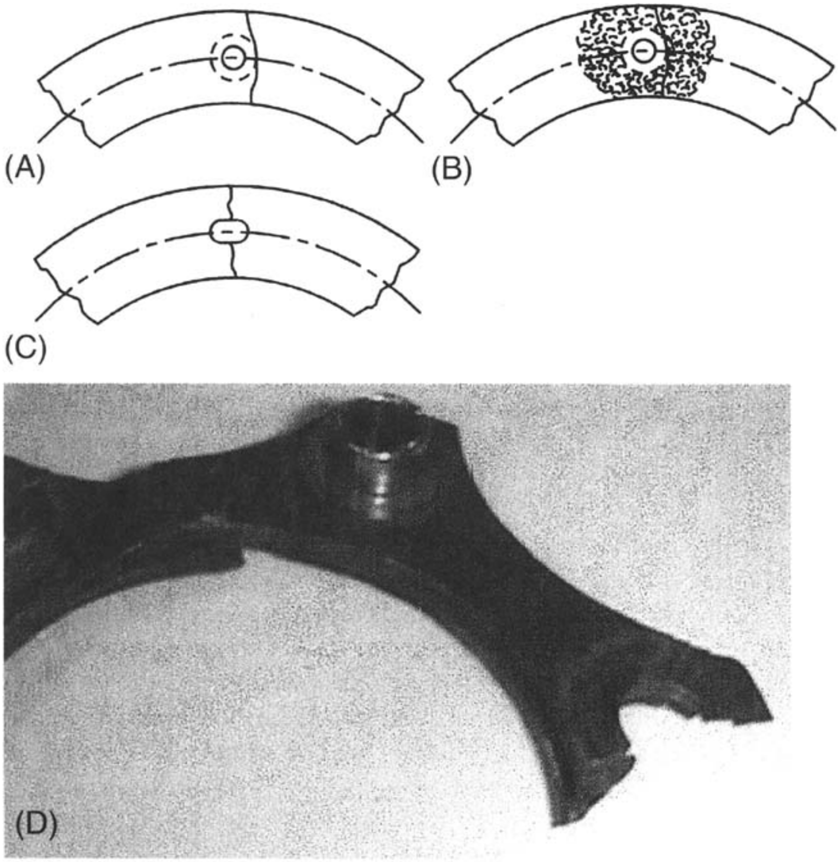


Figure 8.34 Disc failures. (A) Misalignment failure. (B) Fatigue failure. (C) Sketch of elongated bolt holes. (D) Picture of elongated bolt holes.

bolt and tighten the locknut to the proper torque. Do not turn the bolt during the locknut-tightening process.

3. The disc pack is wavy, and the dimension between flange faces is smaller than indicated on the installation instructions or applicable assembly drawing (Figure 8.35A). The coupling has been installed in a compressed condition, or equipment

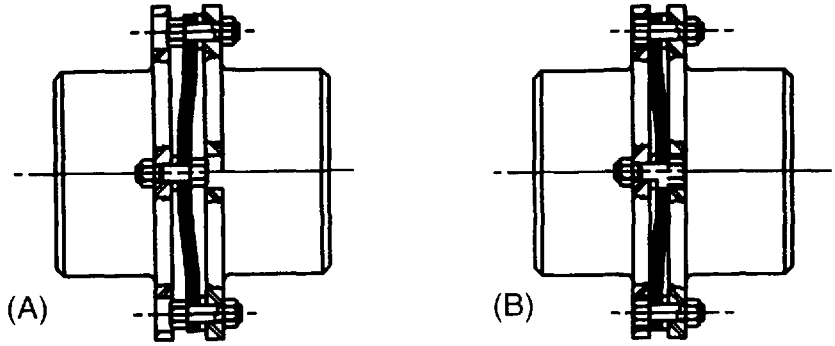


FIGURE 8.35 Distorted disc packs (tin-canned). (A) Compression damage of a disc pack. (B) Elongation damage of a disc pack.

has shifted axially during operation. Check for thermal growth problems. If the application is a sleeve-bearing motor, make sure that the operating centerline of the motor rotor is properly positioned. Make necessary adjustments to relieve disc pack compression during operation.

4. The disc pack is wavy, and the dimension between flange faces is larger than specified on the installation instructions or applicable assembly drawing (Figure 8.35B). The coupling has been installed in an elongated position, or equipment has shifted axially during operation. Realign the axial position of the equipment so that the coupling operates with a neutral flat disc pack. If the application is a sleeve-bearing motor, make sure that the operating centerline on the motor rotor is in the proper position.
5. The disc pack has a bulge near the center or is bowed toward one flange in alternate chord positions, or the bolts are bent (Figure 8.36). This condition is a result of a large torque overload induced in the system above the peak overload capacity of the coupling. The remaining disc pack chordal sections will be very straight and tight. Check the coupling selection; apply the proper size and style for the characteristics present.

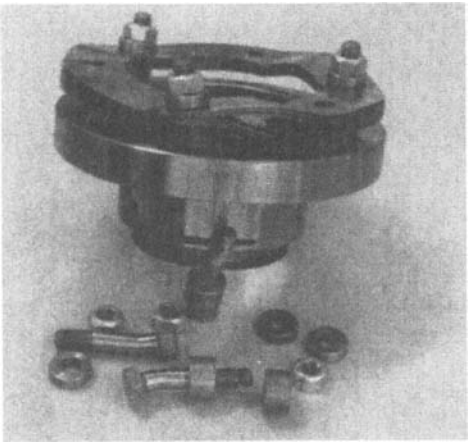


FIGURE 8.36 Torque-overload failure.

III. DIAPHRAGM COUPLINGS

The flexible diaphragm coupling is available in three basic forms (Figure 8.37). This coupling obtains its flexibility from the free span between the diaphragm's outer and inner diameters. Torque for these couplings is transmitted between the O.D. and the I.D. The diaphragm elements can be of constant or variable thickness, usually with maximum thickness at the smaller diameter. All three shapes commonly used have some shape modification that helps to reduce their size and increase their flexibility.

The tapered contoured diaphragm (Figure 8.37A) coupling has as its flexible element a thin profile diaphragm machined from a solid disc of heat-treated alloy material. This diaphragm is contoured so that it has a nearly uniform torsional shear stress through the profile, which is therefore thicker at the hub, or I.D., and progressively thinner at the rim, or O.D. The purpose of contouring the profile is to keep the diaphragm as thin as possible consistent with the transmitted torque. This keeps the misalignment bending and axial stresses as low as possible for a given torque. (See Figure 8.38.)

The thickness of a diaphragm can be changed to permit a trade-off between torque and flexibility. A thicker diaphragm has greater torque

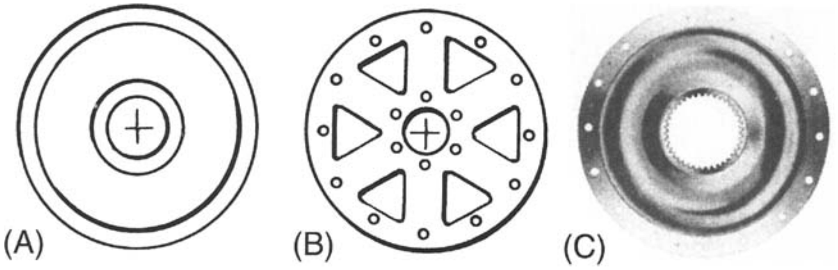


Figure 8.37 Types of flexible diaphragm couplings. (A) Tapered coutoured diaphragm (courtesy of Lucas). (B) Multiple straight diaphragm (courtesy of Flexibox International, Metastream Couplings). (C) Multiple convoluted diaphragm (courtesy of Ameridrives International).

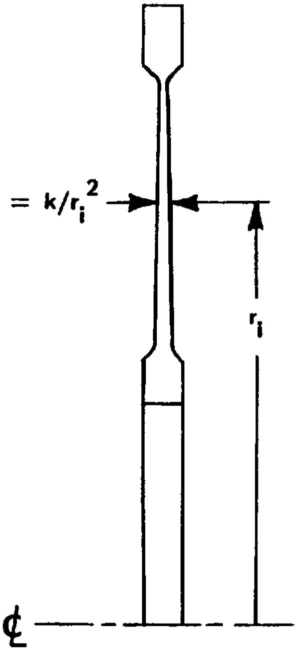


Figure 8.38 Typical shape of contoured diaphragm.

capacity but is not as flexible, and vice versa. The reason for machining the diaphragm from a solid disc was to provide a smooth fillet junction between the flexing portion and the rigid integral rims and hub that connect to the rest of the coupling. Today with the use of finite element analysis (FEA), diaphragm profiles can be optimized to profile the "optimum" in stresses for application-specific needs.

There are two type of flexible element attachments. The most often used "marine style" coupling configuration allows the mounting hub bore to vary considerably without affecting the diaphragm diameter (Figure 8.39). Thus, the coupling is sized to fit the torque and misalignment requirements rather than having its size dictated by the connected machine shaft size. For this design the diaphragm can be electron beam welded to the spacer ("electron beam design", Figure 8.40) or with today's sophisticated computer numerical control (CNC) equipment they can be machined with an integral I.D. machined, flanged, "one-piece design" (Figure 8.41).

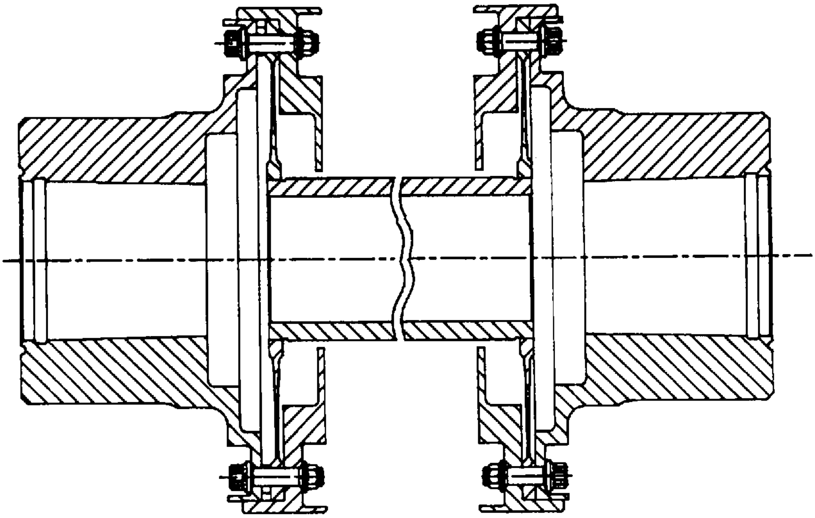


Figure 8.39 "Marine style" contoured diaphragm coupling.

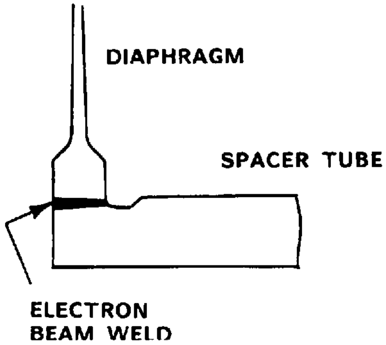


FIGURE 8.40 Electron beam welded diaphragm.

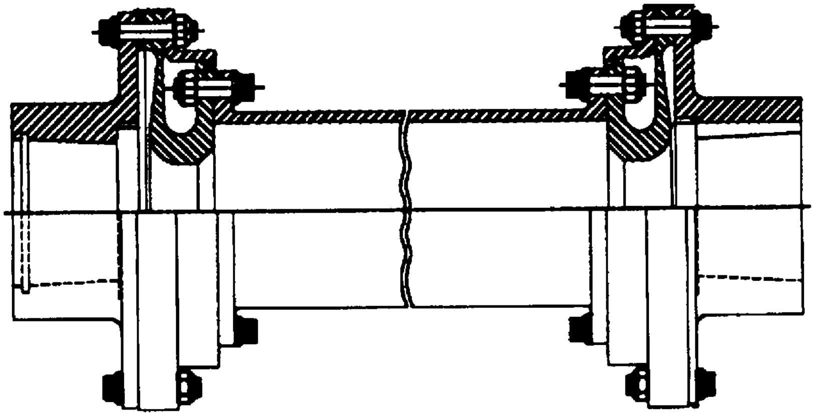


FIGURE 8.41 One-piece coutoured diaphragm. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

For those applications requiring a reduced moment coupling, the contoured diaphragm is machined integral with the hub (Figure 8.42). For the one-piece design, the diaphragm O.D./I.D. ratio is reduced and turned inside out so it can be mounted over the hub (Figure 8.43). These configurations shift the flexible center closer to the machinery bearings to reduce the overhung moment from the weight by moving the over-the-coupling center of gravity (C.G.) toward the machine bearings.

The multiple straight diaphragm (with spokes) (Figure 8.37B) has

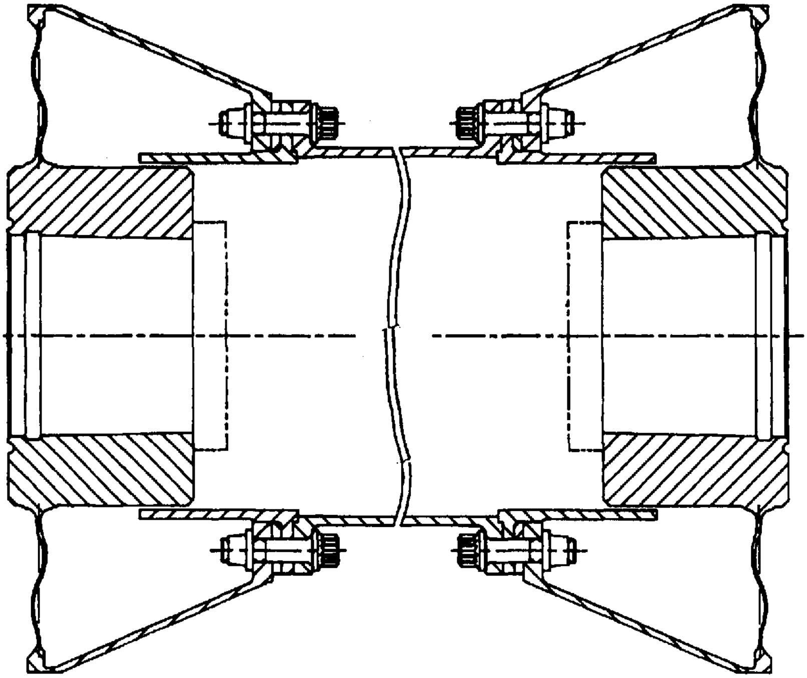


FIGURE 8.42 Reduced moment coutoured welded diaphragm. (Courtesy of Lucas.)

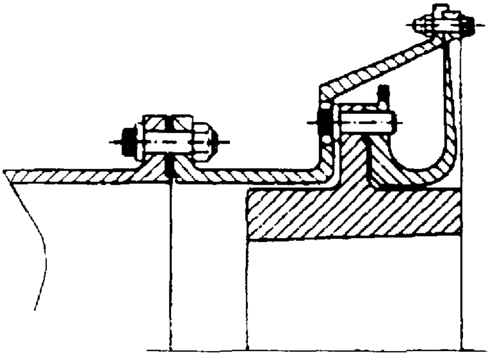


FIGURE 8.43 Reduced moment one-piece contoured diaphragm. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

multiple thin plates rather than one thick plate, which greatly increases the flexibility of this couple compared to that of a single diaphragm. With thin diaphragms in parallel, the stress is usually lower. The stresses, moments, and forces of a diaphragm increase with the third power of the thickness (T^3). Hence the use of several thinner diaphragms produces lower values (Nt^3), where Nt is equal to T for a single diaphragm coupling. The coupling shown in Figure 8.37B also has material removed (forming spokes) to increase flexibility further.

The multiple convoluted diaphragm (Figure 8.37C) incorporates all the flexibility of the one shown in Figure 8.37B but also has a convoluted shape, which helps to increase its flexibility in the axial direction and provides linear axial stiffness (refer to Figure 8.7).

A. Construction of Diaphragm Couplings

The diaphragm coupling is usually used on high-performance equipment and therefore must have high reliability. Because of this, its torque-transmission components are usually made of high-strength alloys such as AISI 4140 and 4340.

Flexing diaphragms are made from high-strength alloys that exhibit good fatigue properties. Tapered contoured diaphragms are usually made from AISI 4100 or 4300 steels coated for corrosion protection. Multiple diaphragms (straight and convoluted) are typically made from cold/reduced 300 Series stainless steel (1/4 to 1/2 hard condition), may use some of the PH stainless steels (15-5PH, 17-4PH, or 17-7PH), or may be made from some of the high-strength nickel alloys (such as 718).

B. Applications of Diaphragm Couplings

The diaphragm coupling is typically used in high-performance applications where a failure or downtime can be costly (see Figures 8.44 and 8.45).

C. Types of Diaphragm Couplings

There are many types of diaphragm couplings, including

1. Tapered contoured diaphragm (Figure 8.46)

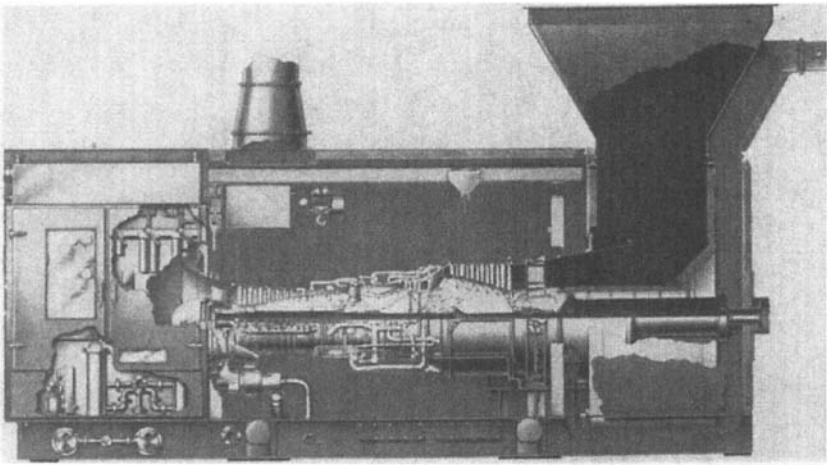


FIGURE 8.44 Industrial gas turbine with double-tapered contoured coupling. (Courtesy of General Electric Schenectady, Gas Turbine Division.)

FIGURE 8.45 Accessory drive diaphragm coupling. (Courtesy of Ameridrives International.)

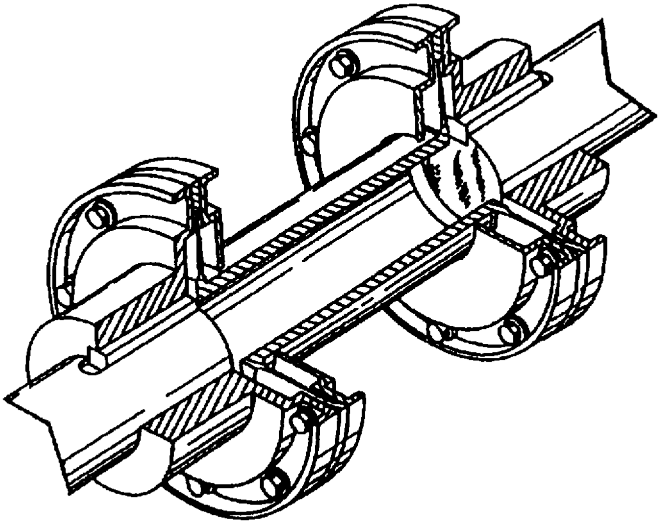


FIGURE 8.46 Tapered contoured diaphragm coupling.

2. Multiple straight-spoke diaphragm (Figure 8.47)
3. Multiple convoluted diaphragm (Figure 8.48)
4. One-piece contoured diaphragm coupling (gas turbine coupling) (Figure 8.49)
5. Reduced moment multiple convoluted diaphragm (Figure 8.50)

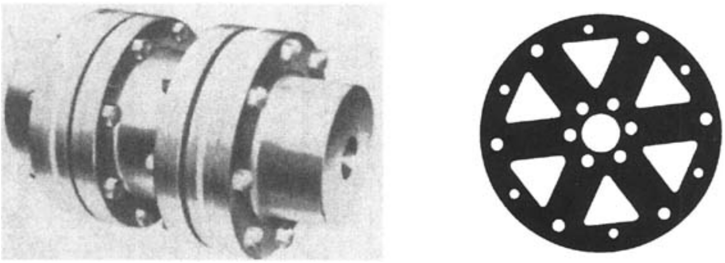


FIGURE 8.47 Multiple straight diaphragm coupling. (Courtesy of Flexibox International, Metastream Couplings.)

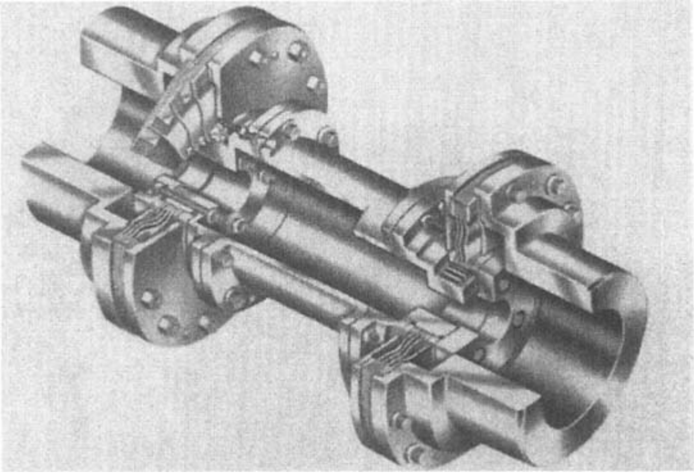


FIGURE 8.48 Multiple convoluted diaphragm coupling. (Courtesy of Ameridrives International.)

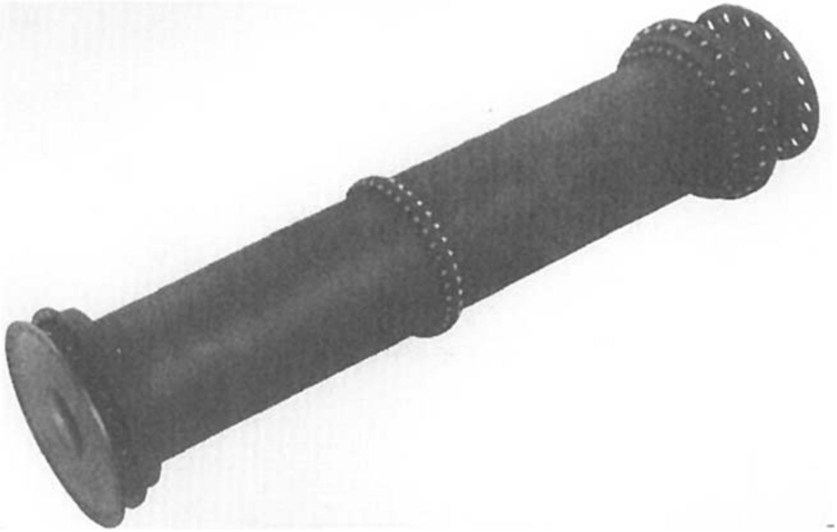


FIGURE 8.49 One-piece contoured diaphragm. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

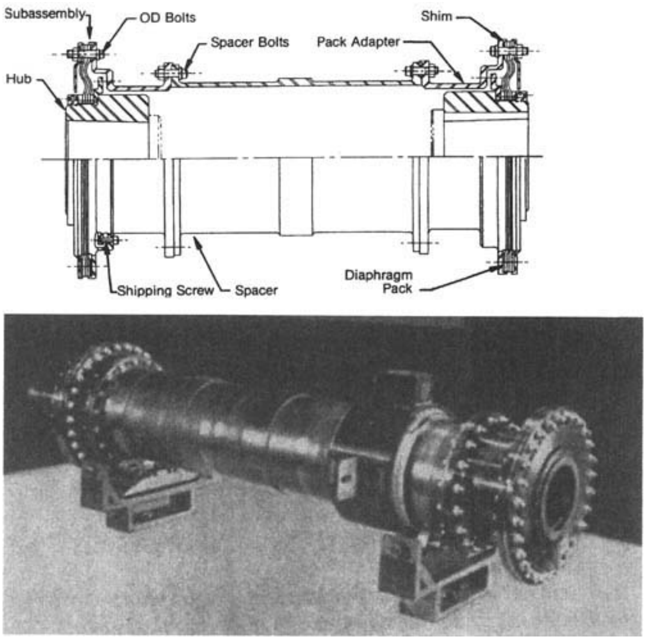


Figure 8.50 Reduced moment multiple convoluted diaphragm. (Courtesy of Ameridrives International.)

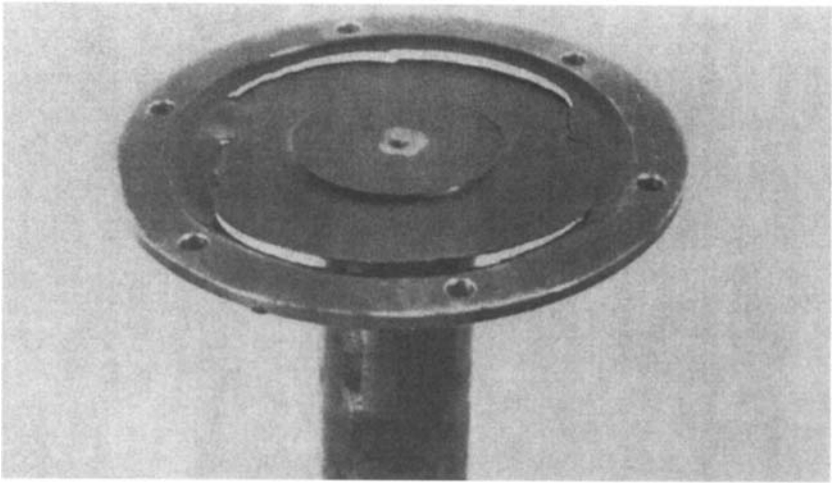


Figure 8.51 Failure of a tapered contoured diaphragm coupling. (From Michael Neale et al., Proceedings of the International Conference on Flexible Couplings for High Power and Speed, 1977.)

D. Failure Modes for Diaphragm Couplings

Some of the typical failure modes for the three basic types of diaphragm couplings are as follows:

Typical failure of tapered contoured diaphragm from over-misalignment and excessive axial travel (Figure 8.51)

Failure of a diaphragm coupling from overtorque (Figure 8.52)

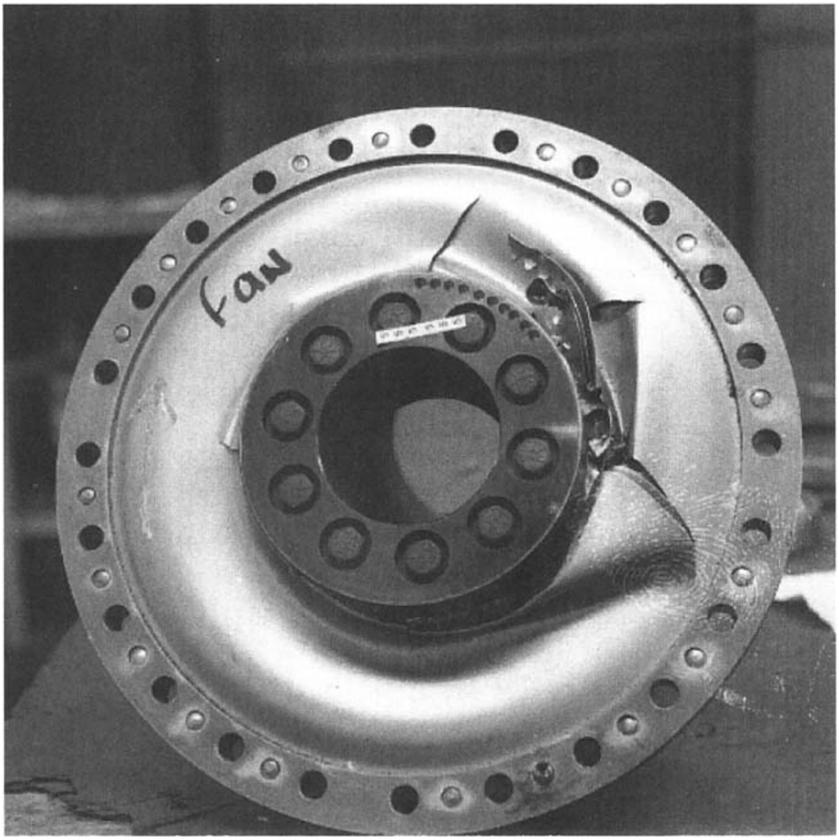


FIGURE 8.52 Failure of a diaphragm coupling from overtorque. (Courtesy of Ameridrives International.)

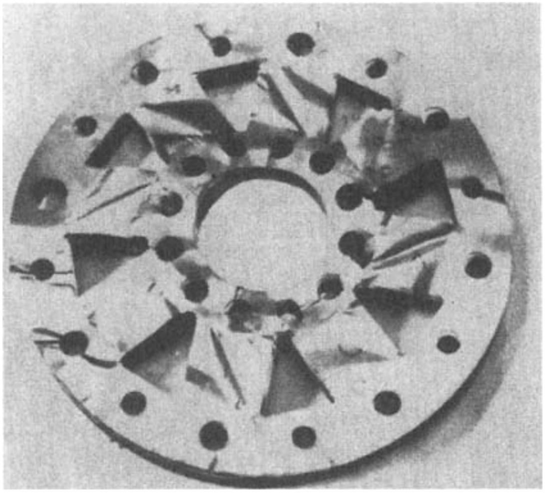


FIGURE 8.53 Failure of a multiple diaphragm coupling due to overtorque. (From Michael Neale et al., Proceedings of the International Conference on Flexible Coupling for High Power and Speed, 1977.)

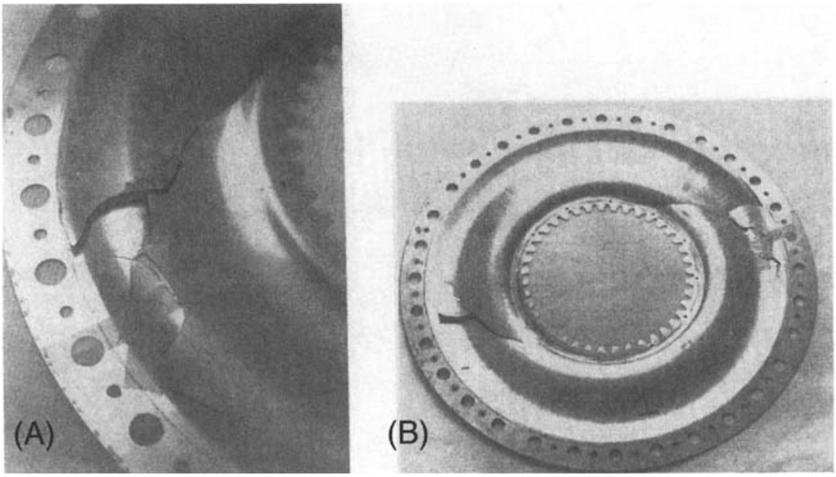


FIGURE 8.54 Stress corrosion fatigue failure of a diaphragm. (A) Failed outer diaphragm. (B) Failed outer diaphragm with corrosion pitting. (Courtesy of Ameridrives International.)

Multiple straight-diaphragm failure from overtorque (Figure 8.53)
Multiple convoluted diaphragm coupling failure: stress corrosion fatigue (Figure 8.54)
Failed outer diaphragm (Figure 8.54A)
Failed outer diaphragm with corrosion pitting (Figure 8.54B).

This page intentionally left blank

9

Miscellaneous Couplings

I. INTRODUCTION

Chapters 5–8 covered some of the most commonly used couplings and U-joints. There are many other types available. This group of flexible couplings obtains flexibility from a combination of the mechanisms discussed in Chapters 6–8 or a unique mechanism. Some of these are

1. The pin-and-bushing coupling (Figure 9.1)
2. Offset coupling (Schmidt coupling) (Figure 9.2)
3. Tangential spring couplings (Figure 9.3)
4. Metallic beam coupling (Figure 9.4)
5. Sliding block coupling (Figure 9.5)

In this chapter we describe their operating principles and functions and discuss the variations available.

II. PIN-AND-BUSHING COUPLING

In a pin-and-bushing coupling (Figure 9.1A), resilient drive is provided by rubber-cushioned sleeve bearings. Misalignment is accommodated by compression of the rubber bushing. End float is accommodated by the sliding of hardened-steel drive pins in the bronze inside diameter on the rubber bushing. No lubrication is required. This type of coupling not only transmits torque and accommodates for misalignment and axial travel but also absorbs small amounts of torsional vibration. Pin-and-bushing couplings are generally used on small-horsepower applications such as pumps, hoists, conveyors, and paper converting equipment.

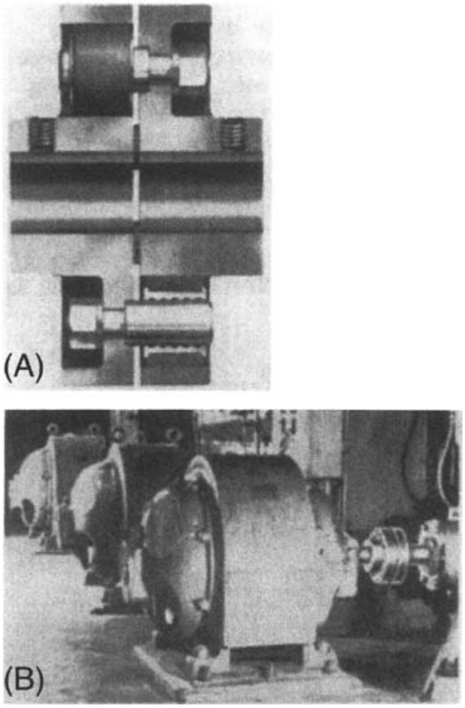


FIGURE 9.1 (A) Pin-and-bushing coupling. (B) An application of the pin-and-bushing coupling. (Courtesy of Renold, Inc., Engineered Products Division.)

Misalignment capacity, angular up to 1° and offset up to 0.003 in. times the coupling outside diameter. Couplings range from 4 to 15 in. with torque capacities to 125,000 in.-lb.

III. OFFSET COUPLING (SCHMIDT COUPLING)

The offset or Schmidt coupling (Figure 9.2) belongs in the family of torque rigid couplings and is designed to provide for large parallel shaft offset without side loads or secondary forces on the equipment. Torque is transmitted through a series of plates by mechanical linkages while allowing great translation of the rotating axis of each of the plates. These couplings can handle from 3/16 to 17 in. of offset with torque capacities of 55–460,000 in.-lb In-lbs of torque. They are used on steel mill equipment, conveyors, etc.

IV. TANGENTIAL SPRING COUPLING

Tangential spring couplings (Figure 9.3) transmit torque through tangentially arranged compression springs, need no lubrication, and are unaffected by most chemicals. The spring rate and inertia can be varied. The standard design permits maximum torsional deflections of approximately ±5°. Special designs are available for deflections between ±1°

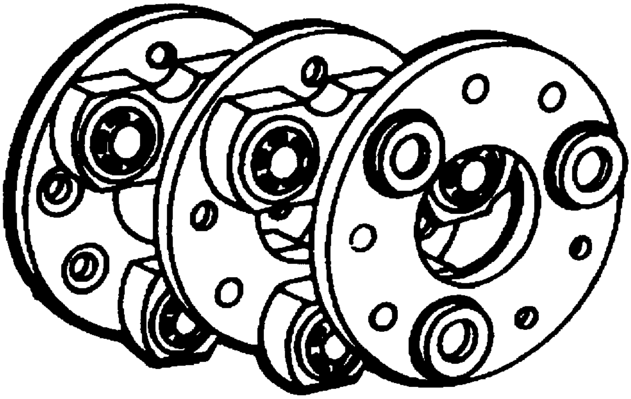


FIGURE 9.2 Offset coupling. (Courtesy of Zero-Max, Motion Control Products.)

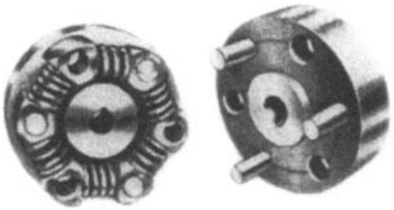


FIGURE 9.3 Tangential spring coupling. (Courtesy of Renold, Inc., Engineered Products Division.)

and ±8°. The spring is available in sizes to transmit more than 50,000 hp and up to 15×10^6 in.-lb of torque. The spring coupling is also available with V-pulleys, flywheels, and special adapters.

Spring couplings protect connected equipment against expensive downtime by cushioning shock loads. These couplings will accommodate shaft misalignment and axial travel, both singly and in combination. When combined misalignments occur, permissible individual misalignments are reduced.

V. METALLIC BEAM COUPLING

The metallic beam coupling (Figure 9.4) is a curved beam that transmits torque and accommodates misalignment and end movement with no backlash and with constant velocity. The one-piece coupling is manu-

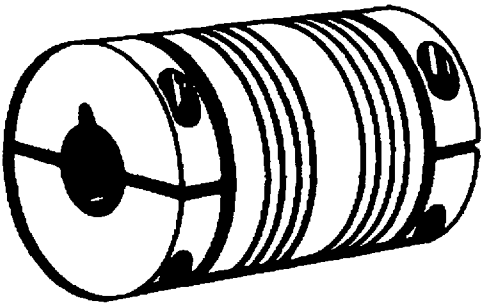


FIGURE 9.4 Metallic beam coupling. (Courtesy of Helical Products Co., Inc.)

factured from such metals as aluminum alloys or stainless steels (including precipitated hardened steels such as 17-7 PH). The coupling resistance to torsional deflection allows it to handle rated torque capacities with less than 7° of windup. Its resistance to shear deflection allows two flexible units to support a long driveshaft and maintain dynamic stability. The beams are shaped to maintain equal stress at the inside and outside diameters, which makes it possible to carry the same torque in either direction of rotation. In industrial applications these couplings have been used on pumps of up to 100 hp at speeds of up to 3600 rpm.

VI. SLIDING BLOCK COUPLING

The sliding block flexible coupling (Figure 9.5) is a simple efficient unit. It performs all the required functions of a flexible coupling and compensates for angular misalignment up to ±1°. It is used for blind assembly or vertical applications.

The sliding block coupling is generally known as the "Oldham coupling." It transmits torque through an intermediate square floating member and compensates for all three types of misalignment by the combined sliding action between the closely fitted center member and the adjacent driving and driven jaw flanges.

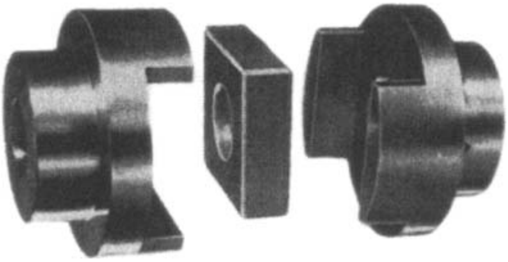


FIGURE 9.5 Slider block coupling. (Courtesy of Ameridrives International.)

This page intentionally left blank

10

Special-Purpose Couplings

I. TORQUEMETER COUPLINGS

A. General

The use of continuous torque monitoring couplings is quickly becoming an integral part of many predictive maintenance programs in the petrochemical and process industries. More and more operating facilities are using these instrumented torque-measuring couplings in order to know how their critical equipment is performing so that intervals between scheduled shutdowns can be chosen appropriately. Also, when energy costs rise, competitive pressures demand that operating facilities run their equipment as efficiently as possible, adding to the popularity of these so-called torquemeter (or torsionmeter) couplings.

Torquemeters physically measure the power (actually torque and speed) being transmitted between the two machines to which they are connected. The main output of the torquemeter is torque, but usually speed is also measured. This allows the torque and speed to be multiplied for a readout of power. When installed between a gas turbine and compressor, for example, a torquemeter enables the operator to optimize the turbine operation for maximum output and minimum fuel consumption. Torquemeters are often connected directly to the main driver machines for such specific uses as load-sharing optimization of multiple drivers, antisurge control systems, gas turbine–compressor fouling determination, and turbine blade deposit determination. When connected between compressors in multibody trains they have been able to identify the compressor responsible for the overall decrease in energy output.

When dealing with torquemeters, a topic often discussed is accuracy. It is common for this accuracy to be expressed in terms of a percentage of full-scale torque. These limits tend to become ambiguous unless the full-scale torque is also defined. For these reasons it is recommended that the torquemeter manufacturer be given the operating conditions and that accuracy be discussed in terms of inch-pounds to avoid any confusion.

B. Basic Methods of Measurement

All torquemeter designers are faced with the same problem. Each system detects a physical change in the coupling due to torsion while it is rotating and must get the signal to a stationary output device (generally in the control room). Each system is faced with determining those physical changes associated with torque alone while the coupling is subjected to a combination of torque, bending, and centrifugal loads. Discriminating the effects of these multiple loads boils down to two basic methods of detection: (1) measurement of localized torsional strain and (2) measurement of overall torsional deflection.

C. Logistics

The logistics of procuring a torquemeter usually involve communication between the coupling manufacturer and the torquemeter manufacturer. Coordination between these two vendors will usually involve modifications to be made to the coupling spacer (center spool piece). The coupling spacer is sent to the torquemeter vendor for possible modification and calibration of the unit. It is then returned to the coupling manufacturer to be balanced. Finally, the entire coupling with all the final electronics is shipped to the installation site for final assembly. Usually a field service technician is required to help with the final installation. This activity must be well communicated ahead of time so that the delivery schedule can be planned accordingly by all parties.

D. Strain Gage Types of Torquemeters

There are several variations of the strain gage torquemeter system. Each of them operates on the same general principle of

1. Getting electrical operating power from an outside source to the coupling
2. Feeding that power through a four-arm strain gage bridge located on the rotating coupling
3. Transmitting the resulting signal from the coupling back to a stationary receiver.

The strain gages are usually directly affixed to either the outer or inner diameter of a thinned-down area on the coupling spacer (center spool piece). (For lower speed applications, some manufacturers provide a clamp-on split collar that contains the strain gages.) As torque is applied, the localized twisting in the area of the strain gages creates a signal by unbalancing the strain gage bridge. Since the coupling spacer will be exposed to axial, centrifugal, and misalignment loads in addition to torque, the strain gages of the Wheatstone bridge (see Figure 10.1) must be mounted precisely at 45° from the coupling's axis in order to minimize the strains from these extraneous loads.

In the past, the method of transmitting power to and receiving signals from the rotating instrumented torque-measuring coupling included the use of contacting slip-ring arrangements. This rendered them useful for only low-speed, high-torque applications and presented problems related to wear and foreign particulates. Today, most strain gage torquemeter systems have overcome these problems by using noncontacting electromagnetic induction techniques.

The basic strain gage type of torquemeter (see Figure 10.2) con-

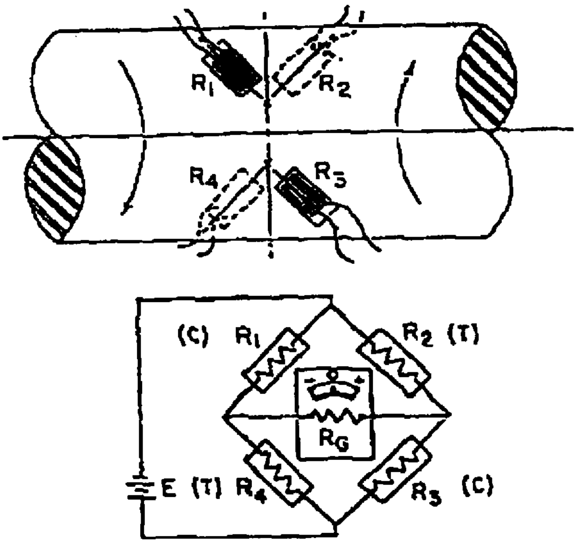


Figure 10.1 Typical Wheatstone bridge used in a strain gage torquemeter.

sists of a stationary component and a rotating component, both of which contain electronics. The stationary component (stator) provides power to the rotating component (rotor) via electromagnetic induction. The air gap between the stationary and rotating windings allows for relative axial, angular, and offset excursions of the coupling during operation.

The rotating electronics condition the signal received from the stationary component and feed it through the strain gage circuitry. (The rotating strain gage circuitry is usually provided by the manufacturer with a protective wrapping due to its sensitivity to damage from handling and possible chemical contaminants.) Since the strain gage signal levels are typically low, they require careful processing to avoid noise- and drift-related errors. The output of the rotating strain gage circuitry is amplified and transmitted back to the stationary component either by a frequency-modulated (FM) signal or by a second rotary transformer, depending on the manufacturer.

From the stationary component, the signal is typically transmitted

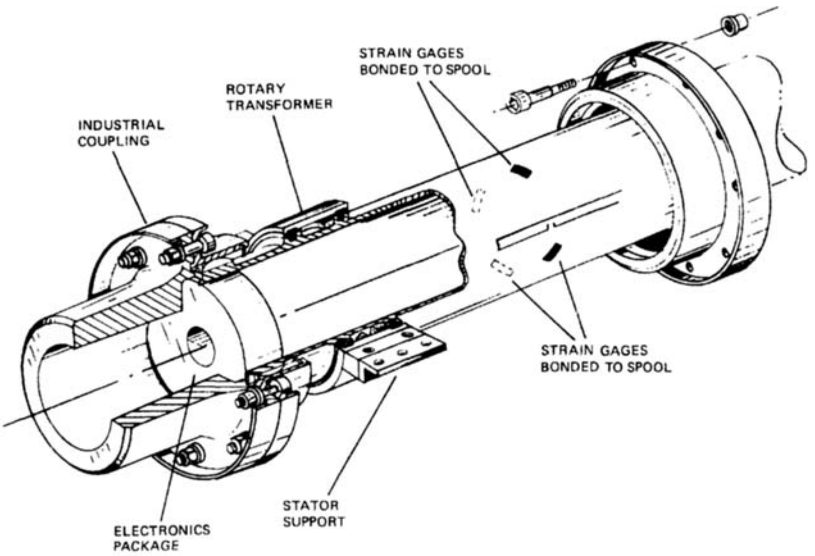


Figure 10.2 Typical strain gage torquemeter coupling. (Courtesy of Indikon Company.)

to the control room as an industry standard analog signal for the user's data recorders or programmable logic controllers (PLCs).

E. Torsional Deflection Types of Torquemeters

As with stain gage types of torquemeters, there are several variations of the torsional deflection torquemeter system. Each of them operates on the same general principle of measuring the torsional "windup" experienced when the coupling is exposed to torque by comparing the relative circumferential positions of different locations along the coupling's axis. Many methods have been devised to measure the torsional windup exhibited by the coupling. These methods range from measuring changes in the acoustics of coupling-mounted piano wires to the application of magnetic circuits that sense changes in permeability as the coupling windsup. The most practical way of measuring the coupling torsional windup has been found to be the measurement of the phase shift of separate speed pickups mounted along the coupling's length. In this respect,

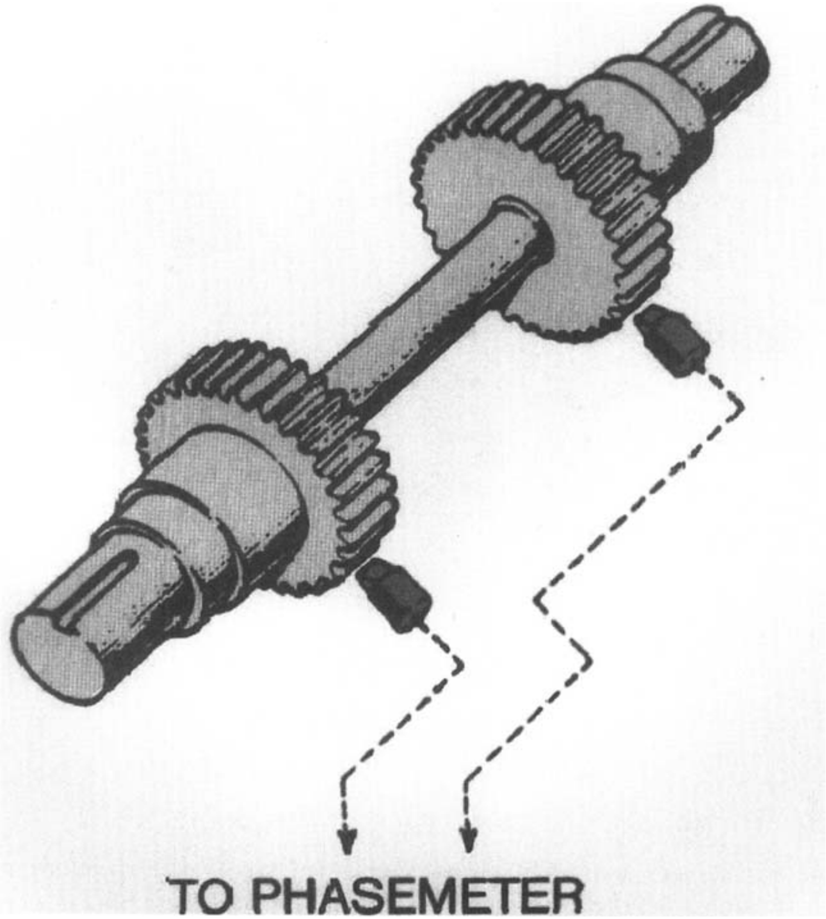


Figure 10.3 Torsional deflection torquemeter.

torsional deflection torquemeters have become synonymous with *phase-shift* torquemeters (see Figure 10.3).

Each variation of the *phase-shift*-measuring *torsional defection* type of torquemeter senses the relative positions of opposite ends of the coupling using a pair of toothed flanges that are an integral part of the coupling's spacer. The sensing devices are stationary and work on the same principle as a typical speed pickup, where the flux field around the pickups are changed every time a (steel) tooth on the rotating coupling passes it. By monitoring the phase relationship between toothed wheels affixed to each end of the coupling, an indication of coupling "twist" is obtained.

Unfortunately, vertical and horizontal movements of the rotating coupling relative to the nonrotating pickups will also produce a phase shift. The methods of distinguishing between torsion-induced phase shift and phase shifts caused by these lateral movements form the basic differences between torsional deflection torquemeter systems.

Another common characteristic of all phase-shift torsional deflection torquemeters is that they are not nearly as sensitive to tooth manufacturing tolerances as one might think. The actual tooth movement at full-scale torque for a typical application is usually less than 1/16 in., so a typical accuracy of ±1% requires that the tooth positions be known (relative to each other) to within 1% of that, which relates to values far less than 0.001 in. A typical misconception is that each tooth has to be manufactured to locational tolerances far less than 0.001 in. in order to "know" where each tooth is positioned. In actuality, the location of every tooth is not required to be held to such excruciating tolerances because of the way the units are calibrated. The calibration techniques involve "knowing" where the *average* tooth position is for the entire set of teeth on each toothed wheel as the coupling spins at speed. For this type of torquemeter the voltage signals sensed by the pickups are typically sent to the control room, where they are processed. The output of the processing unit is also typically made available as an industry standard analog signal for connection to the user's data recorders or PLCs. All phase-shift torsional deflection torquemeters measure coupling rpm as a by-product of determining torque, allowing the torque and speed signals to be multiplied for a direct readout of power.

1. The Two-Channel Phase-Shift System

One manufacturer handles the problem of distinguishing genuine twist-induced phase shifts from false induced phase shifts (produced by lateral movements at either end of the coupling) by providing a version of pickup that senses the position of the toothed wheels around many points along the circumference of the coupling. Rather than supplying an array of circumferentially mounted speed pickups (with multiple

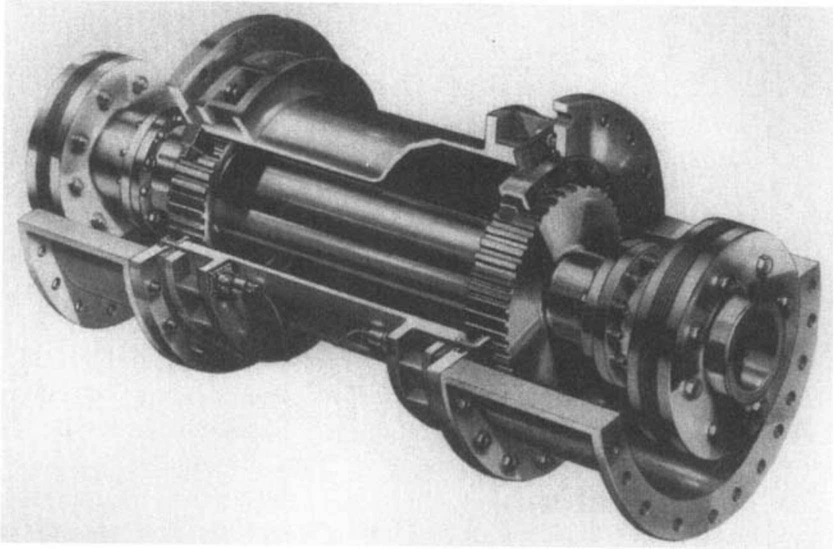


FIGURE 10.4 Pickup rings used in a two-channel system.

cabling error sources), each pickup takes the form of an internally toothed ring. With only two pickup rings, there are only two cabling error sources. Each of these two pickup rings (Figure 10.4) completely surrounds each rotating toothed wheel of the coupling, and each consists of circumferentially wound coils, which are then energized to create a toroidal flux path. As the coupling *lifts* on one end relative to that pickup ring (mounted in a special housing), the *early* signal on the front side of the coupling is therefore canceled by a *late* signal at the back side of the same toothed wheel. In this way, as the average position of teeth (on, say, the driving side of the coupling) cuts through the flux field of the driving side pickup ring (say channel A), a sinusoidal voltage is created and sent to the processing unit (typically located in the control room). Similarly, a sinusoidal voltage is sent to the processing unit on channel B for the average tooth position on the driven side of the coupling.

The pickup rings used on the two-channel system cannot be horizontally split and therefore must completely surround the toothed portion of the rotating coupling. The horizontally split coupling guard must

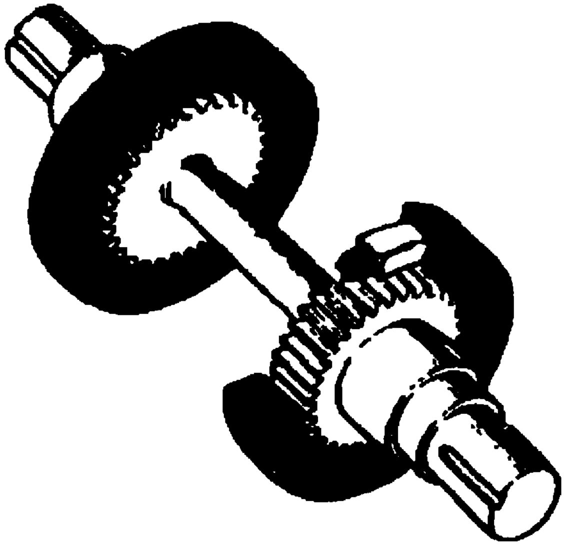


FIGURE 10.5 Integral coupling guards with a two-channel system. (Courtesy of Torquemeters Ltd.)

therefore be replaced with one supplied by the torquemeter manufacturer. The coupling guard becomes an integral part of the torquemeter installation (Figure 10.5).

The processing unit receives the sinusoidal voltages of the averaged position of the driving wheel on channel A and compares them to the similarly averaged location of the driven wheel on channel B. The difference in phase between these two channels is a direct measure of torsional windup deflection. By multiplying this torsional deflection by the predetermined coupling stiffness (measured during initial laboratory calibration), the processing unit is then able to determine the torque present in the coupling. Since the frequency of the sinusoidal voltage signal is simply the coupling rpm times the number of teeth on each wheel, the processing unit is also able to directly output a reading of the power being transmitted by the coupling.

2. The Three-Channel Phase-Shift System

Another method of deciphering twist-induced phase shifts from false laterally induced ones is to move the two toothed wheels of the coupling closer together to minimize the errors caused by lateral movements of the coupling relative to its housing. Simply shortening the length of tube between toothed wheels would not work, because it would result in a proportionally smaller amount of torsional twist. Instead, the (first) toothed wheel that is to be moved closer to the other (second) toothed wheel is still affixed to the coupling at the same axial location as in the two-channel system, but the manufacturer adds a non-torque-carrying reference sleeve (see Figure 10.6). The function of the reference sleeve is simply to translate the apparent location of the (first) toothed wheel, moving it closer to its paired (second) toothed wheel without moving the location of the attachment point to the torsion tube underneath the sleeve.

In this arrangement, the reference tube is attached to the torque-carrying torsion tube at one end only. The attachment side is at the position where the first wheel *would have been* if it had not moved over to the other side. The reference sleeve carries the first wheel to the other end of the coupling by cantilevering off its fixed end. To eliminate a projected offset of the first wheel as the torsion tube underneath bows from

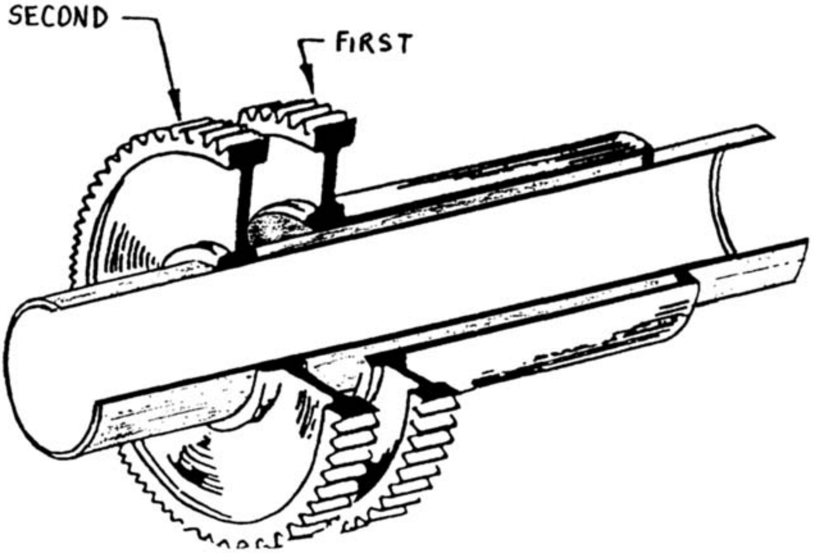


FIGURE 10.6 Two-wheel phase-shift system.

misalignment, the free end is kept from being laterally displaced by means of a radial bearing.

By moving the axial location of the first toothed wheel closer to that of the second toothed wheel, the error of false laterally induced phase displacement is reduced, but since there is still some distance x between these toothed wheels, the error has not been eliminated. This now "smaller" error is canceled by adding a third wheel (see Figure 10.7). The third wheel is positioned exactly the same distance (x) from the second wheel but on the opposite side of it. It is directly attached to the second wheel, so torque is not sensed, only lateral movements. In this way, as the coupling moves laterally within the housing, "earliness" of the signals for the first wheel can be canceled by subtracting the amount of "lateness" of the third wheel.

The sensor design is basically that of a standard variable-reluctance proximity probe; it consists essentially of a wire coil wound around a permanent magnet. As with the two-channel system, as each tooth passes through the flux field of its mating sensor, the processing unit receives a sinusoidal voltage of the average position of the driving (first) wheel on channel A. Similarly, a sinusoidal voltage is received by the processing unit for the average position of the driven (second) wheel

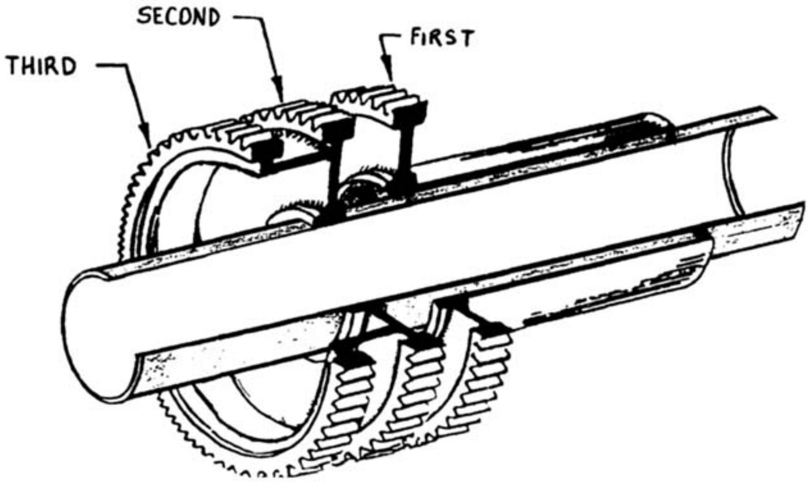


FIGURE 10.7 Three-wheel phase-shift system.

on channel B. Likewise, a sinusoidal voltage is received by the processing unit on channel C for the average tooth position of the compensating reference (third) wheel.

The processing unit (typically located in the control room) receives these sinusoidal voltages of the average location of the three wheels on channels A, B, and C. The phase difference between channels B and C is purely the laterally induced phase-shift error. Knowing this, the processing unit subtracts this error phase difference from the phase difference between channels A and B. This corrected phase shift is then the value associated with torsional windup only.

As with the two-channel system, the torsional deflection is multiplied by the predetermined coupling stiffness (measured during initial laboratory calibration) to give the torque present in the coupling. Since the coupling rpm is once again an indirect result of the frequency of the sinusoidal signal, the processing unit is also able to directly output a reading of the power being transmitted by the coupling.

3. The One-Channel Phase-Shift System

The one-channel system is basically a modification of the three-channel system. The one-channel system combines all three channels into one channel, giving rise to the term "monopole" often used with this type of phase-shift torsional windup torquemeter.

The way in which the one channel is combined is as follows. Like the three-channel system, the one-channel system moves the driving wheel of channel A closer to the driven wheel of channel B. Recognizing that the closer together the two wheels can be placed, the less the error associated with laterally induced phase shift, the wheels are moved so close together that they become *intermeshed* (Figures 10.8A and 10.8B). This feature results in the elimination of the need for the other two channels (B and C), because the intermeshing of the teeth results in sensing the tooth locations at a *single point*. As the coupling moves laterally within its housing, relative lateral displacement of the two wheels has been eliminated. Wherever the coupling moves relative to the sensor, both sets of intermeshed teeth are in the same lateral position.

The problem of determining the phase shift between two separate speed pickups now becomes the determination of phase shift between

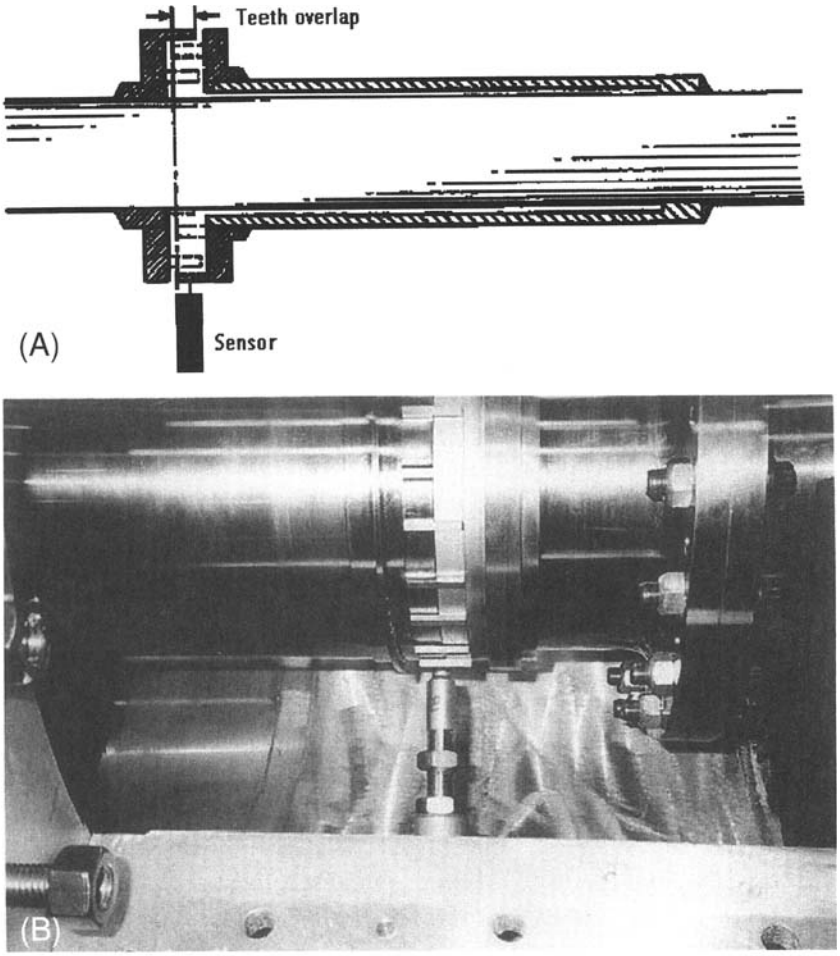


Figure 10.8 (A) Schematic and (B) photograph of intermeshing gear teeth.

consecutive pulses of a single-speed pickup. To illustrate, imagine that the two wheels are intermeshed so that the teeth of the driving wheel are placed equidistant between the adjacent teeth of the driven wheel. As the coupling is rotated under no torque, the single sensing pickup will receive equally spaced increments of voltage pulsations as each tooth on both wheels passes through its flux field. As the coupling experiences torque, the driving teeth will move to an "earlier" location. The resulting signal to the sensor will be unequally spaced increments of voltage. By determining the time lag of these unequally spaced voltage pulsations relative to the equally spaced ones that were seen when the coupling was spun under no torque, a phase shift of consecutive pulsations is established.

This sinusoidal voltage signal is sent to the processing unit (typically located in the control room) where the relative phase shift between consecutive pulses on a single channel is determined as a direct measurement of torsional windup deflection, without the need to correct for laterally induced errors. As with the two-channel and three-channel systems, the torsional deflection is multiplied by the predetermined coupling stiffness (measured during initial laboratory calibration), and the result is the torque present in the coupling. Since the coupling rpm is once again an indirect result of the frequency of the sinusoidal signal, the processing unit is also able to directly output a reading of the power being transmitted by the coupling.

In this design there is usually no radial support bearing at the free end; it is usually cantilevered. As the torsion tube underneath the reference sleeve bows under misalignment, a second independent torque reading is taken on the same set of toothed wheels, but on the opposite side (rear instead of front) of the tooth set (see Figure 10.9). In this way, any "lifting" of the cantilevered wheel will produce an early signal between consecutive pulses in the front of the coupling, with an equally late signal at the rear. Any difference between these two independent phase-shift measurements (each still determined on a single channel) is attributable to laterally induced false phase shifts, while the average phase shift of the two readings results from torsional windup alone.

II. TORSIONAL DAMPING COUPLINGS

A. Machine Vibration

Machine breakdowns often appear to be the fault of the part that failed and are thus attributed to a bad bearing, shaft, or gear. However, the real culprit may be torsional vibration. Machine vibration occurs in three

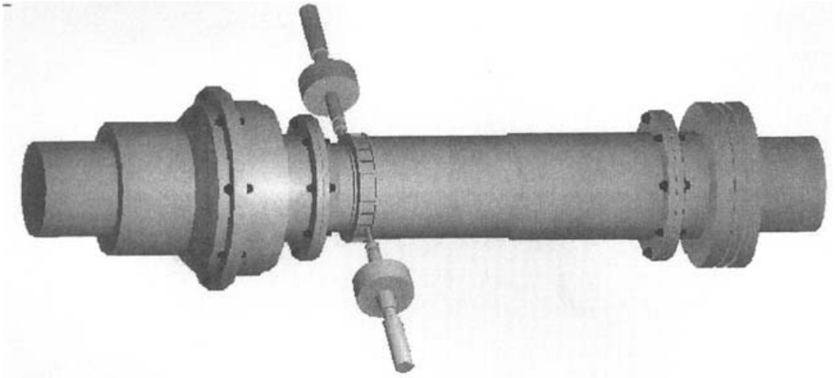


Figure 10.9 Two-pickup system. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

basic forms: lateral, axial, and torsional. Lateral vibration is cyclic displacement perpendicular to the system axis, axial vibration is parallel to the axis, and torsional vibration occurs around the axis.

Of the three basic forms, axial vibration problems in rotating equipment are relatively few. Lateral vibration can damage high-speed equipment because unbalance—the exciting force—is proportional to speed squared. However, torsional vibration poses the greatest potential for serious problems in a wide range of industrial powertrains.

B. Torsional Vibration

Torsional shock and vibration cause various types of failure. If the vibration magnitude is too high for the drive components, failure can occur in one or a few load cycles. This is called *static failure*. At the other end of the spectrum, machines subject to continuous vibration experience over 1 million cycles and must be designed for infinite life to avoid failure. Between these extremes, there are machines that experience a limited number of cycles and are designed for a specific finite life. This concept is illustrated by a fatigue life or *S–N* curve (stress versus number of cycles to failure) (see Figure 10.10).

Torsional failure modes can thus be classified on the basis of the number of stress cycles before failure as follows:

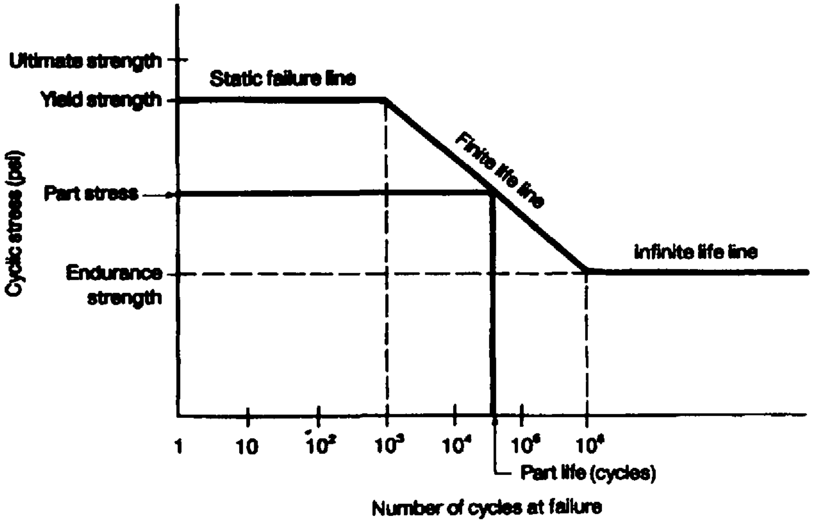


FIGURE 10.10 Fatigue life curve.

1. *Static failure* (less than 1000 cycles). Examples include electric machines subject to electrical faults, centrifugal compressors subject to surges, and synchronous electric motors designed for relatively few starts and stops. Figure 10.11 represents the torque loading on a shaft due to a motor short circuit (fault).
2. *Finite life* (1000–1,000,000 cycles before failure). This includes machines subject to transient vibration at start-up such as synchronous electric motors. Figure 10.12 represents a typical torque loading in this case.
3. *Infinite life* (over 1×10^6 cycles). Examples include machines subject to periodic or continuous vibration such as variable-frequency motors, diesel engines (see Figure 10.13), and reciprocating compressors.

Effective vibration control reduces drive component stresses and increases the number of allowable stress cycles (see Figure 10.10), thereby increasing service life. This stress reduction can also permit the design of lighter parts.

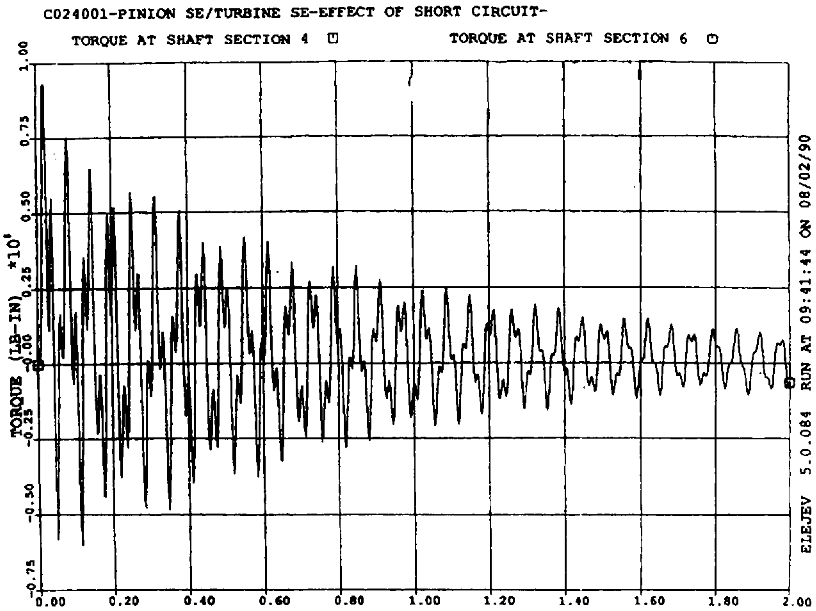


FIGURE 10.11 Torque loading due to motor short circuit.

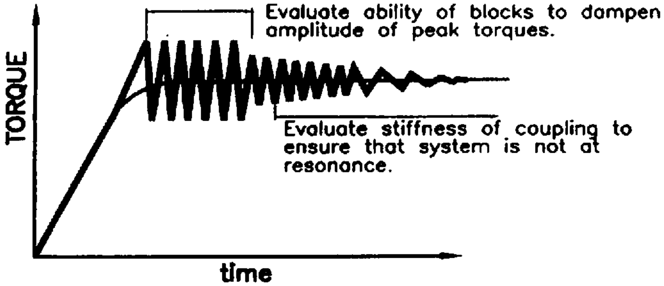

TORQUE
time
Evaluate ability of blocks to dampen amplitude of peak torques.
Evaluate stiffness of coupling to ensure that system is not at resonance.

FIGURE 10.12 Transient vibration at start-up of a synchronous motor.

C. Machine Analysis

Good engineering practice calls for a torsional analysis of large rotating equipment at an early stage in the design so that potentially damaging resonance conditions can be identified. Thus, design changes and damping devices can be evaluated before the design is finalized. Often the

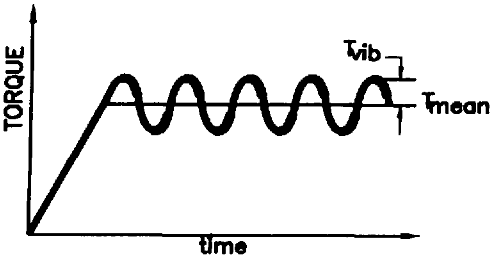


FIGURE 10.13 Periodic or continuous vibration loading of a diesel engine.

design is completed and manufacturing started prior to analysis, leaving little or no opportunity for changes in the major machine components. In this case, the use of a damping device may be the only way to reduce vibration-induced stresses, prevent equipment breakdown, and enhance machine uptime.

TORSIONAL VIBRATION EQUATIONS

There are three basic types of torsional vibration: (1) steady-state, (2) transient vibration when passing through resonance, and (3) transient vibration due to shock load. Typical equations are given for analysis of these vibration types in simple two- or three-mass systems.

Two-mass system with steady-state or transient resonant vibration (see Figure 10.14):

$$F_1 = \frac{60}{2\pi}\left(K_{12}\frac{J_1 + J_2}{J_1 J_2}\right)^{1/2}$$

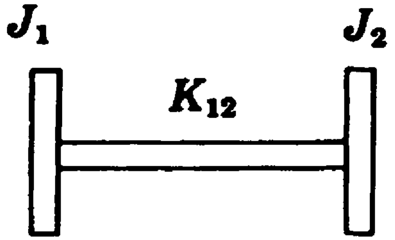


FIGURE 10.14 Two-mass system.

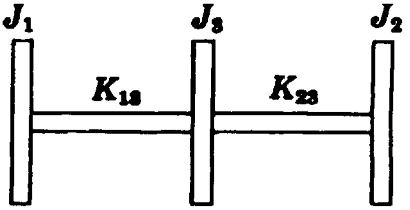


FIGURE 10.15 Three-mass system.

$$T_v = \pm \frac{T_e M J_2}{J_1 + J_2}$$

Three-mass system with transient vibration due to shock load (see Figure 10.15):

$$F_1 = \frac{60}{2\pi}\left\{\frac{a}{2} - \left[\left(\frac{a}{2}\right)^2 - b\right]^{1/2}\right\}^{1/2}$$

$$F_2 = \frac{60}{2\pi}\left\{\frac{a}{2} + \left[\left(\frac{a}{2}\right)^2 - b\right]^{1/2}\right\}^{1/2}$$

$$T_v = T_e\left(\frac{2J_2}{J_1 + J_2 + J_3}\right)\left(\frac{F_2^2}{F_2^2 - F_1^2}\right)$$

$$q = \pm\ 16\, T_v/\pi d^3$$

where

F_1, F_2 = natural frequencies, cycles/min
J_1 = polar moment of inertia of mass to which excitation or shock torque is applied, lb-in.-sec^2
J_2 = polar moment of inertia of load or driver, lb-in.-sec^2
J_3 = polar moment of inertia of intermediate mass, lb-in.-sec^2
K_{12} = torsional stiffness of shaft and coupling between masses 1 and 2, lb-in./rad
K_{13} = torsional stiffness of shaft and coupling between masses 1 and 3, lb-in./rad
K_{23} = torsional stiffness of shaft and coupling between masses 2 and 3, lb-in./rad

M = dynamic magnifier, a nondimensional measure of damping

T_e = excitation or shock torque, lb-in.

T_v = vibratory or peak cyclic torque, lb-in.

$$a = \frac{K_{13}}{J_1} + \frac{K_{23}}{J_2} + \frac{K_{12} + K_{23}}{J_3}$$

$$b = \frac{K_{13}}{J_1}\left(\frac{K_{23}}{J_2}\right)\left(1 + \frac{J_1 + J_2}{J_3}\right)$$

d = shaft diameter, in.

q = cyclic torsional stress in shaft, lb/in.^{2}

If the system has more than three masses, the calculations get more laborious. Alternative approaches are then used such as the Holzer table method with a suitable computer program.

Various devices are used commercially to dampen torsional vibration:

1. Viscous dampers are used primarily for diesel engine vibrations.
2. Friction dampers have limited use because friction discs require frequent adjustment or replacement, and additional couplings are required to handle misalignment.
3. Oil dashpot dampers are used mainly in diesel drives.
4. Damper couplings are used extensively in most types of equipment that suffer from torsional vibration. They dampen lower "driven machine" modes of vibration and can accommodate shaft misalignment.

D. Damper Couplings

Damper couplings fall into two basic categories:

1. Metal spring couplings that employ oil or grease as a damping medium
2. Rubber couplings that rely on hysteresis (internal friction) damping within the rubber elements

The level of damping is about the same with either type of damper coupling. For synchronous motor applications, coupling dynamic magnifiers in both cases are in the range of 3–5.

Rubber damper couplings reduce torsional vibration in two ways. Their soft torsional stiffness reduces the lower natural frequencies below the operating speed range to avoid or minimize resonance, and the rubber dampens the vibration. These couplings also accommodate axial, radial, and angular shaft misalignment between connected machine components.

One type of rubber coupling consists of three main parts: outer sleeve, inner hub, and rubber blocks (see Figure 10.16). Torsional motion between the outer sleeve and the inner hub compresses the rubber blocks and provides damping. Various coupling designs and sizes, as well as rubber compounds and hardnesses, provide a wide range of flexibility and damping.

The method of selecting a coupling depends on whether a system vibration analysis was made (sometimes it is not economically justified).

With an analysis: The design and rubber compound are selected based on the analysis. Peak, normal, and vibratory torque data are used as a basis for selecting the coupling size.

Without an analysis: The design and compound are selected based on experience with the type of drive. For example, diesel engines generally require a torsionally soft coupling, which permits 5–7° of angular displacement at peak torque. For other types of drives, a stiffer rubber coupling is often more effective.

Size is generally based on the required capacity in horsepower per 100 rpm, multiplied by a service factor for the type of prime mover and driven machine. These service factors are typically found in the coupling manufacturer's selection guide.

1. Cost Considerations

A rubber coupling generally costs two to three times as much as a gear-type coupling. However, this often represents only a 1–2% increase in total drive cost.

In a typical case where stiff, nondamping couplings are used, drive components may have to be doubled in strength to handle the higher

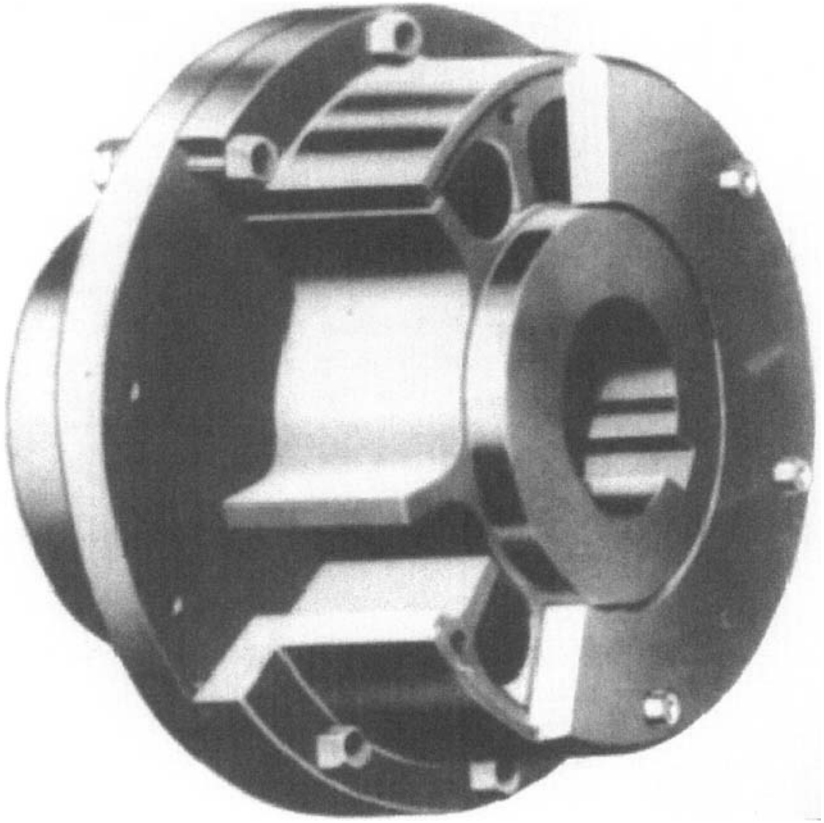


Figure 10.16 Rubber block coupling. (Courtesy of Kop-Flex, Inc., Emerson Power Transmission Corporation.)

peak and vibratory torques. Figure 10.17 is a depiction of the relative coupling torques for the start-up of a synchronous motor, with and without a damper coupling.

2. Typical Application—Variable-Speed Drive

In recent years, variable-speed drives using ac motors controlled by load-commutated inverters (LCIs) have become popular. A typical LCI induces six pulses for each electrical output cycle. Thus, for a 0–60 Hz motor, the pulse frequency varies from 0 to 360 Hz. These pulses can

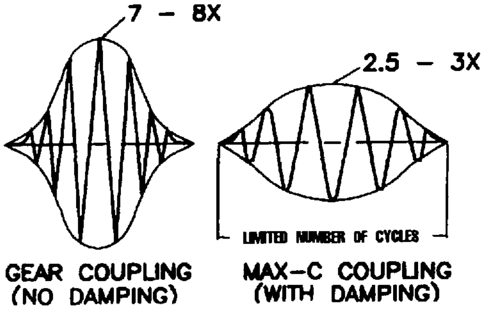


FIGURE 10.17 Typical start-up torques with and without damping.

produce cyclic torques in resonance with one or more system torsional natural frequencies. If resonance occurs within the operating speed range, vibration amplitude builds to a steady-state level; if below minimum speed, it is transient.

An analysis of a 1500 hp variable-frequency motor driving a fan showed a line-to-line electrical fault causing a peak torque 22 times normal full-load torque [22 per unit (p.u.)] with a nondamping coupling. The transient resonant torque was 18 p.u. A rubber damper coupling reduced these values to 11.3 and 2 p.u., respectively.

3. Typical Application—Synchronous Motor

It is now well established that synchronous motors can produce transient torsional vibration during start-up of a magnitude sufficient to cause damage or failure of one or more drivetrain components (see Figure 10.10). The peak vibratory torque developed may cause yielding or failure during the first start. At some lower level of vibratory torque, the cumulative effect over a number of motor starts may eventually culminate in fatigue failure.

The evaluation of the magnitude of vibratory torque and stress is complex. In synchronous motor drives the complexity is accentuated because the motor cyclic torque characteristics are sometimes not well defined. It will be shown later that the vibratory torques and stresses in the drive system are directly proportional to this motor excitation torque.

There are a number of approaches that may be taken to arrive at an acceptable solution. An obvious one is to substantially increase the size of shafting and other torque-transmitting elements to reduce stress to low levels however great the vibratory torque. This solution is not viable with modern high-power, high-speed machinery. Another approach is to employ reduced voltage starting, which reduces the motor's cyclic excitation torque. But this produces a lower system acceleration rate, which permits the buildup of resonant vibration to higher levels, so the net reduction in peak torque levels is often small. A third is to change the inertias and torsional stiffnesses of the system, within the bounds set by other design criteria. Finally, damping can be introduced to reduce vibratory torque to acceptable levels.

The only practical solution is a combination of the last two alternatives—that is, changing torsional stiffness and introducing damping of major proportion. This can be achieved at a cost of roughly 1% of the cost of the drivetrain. An often overlooked, and perhaps the greatest, benefit of this solution is that damping levels can be such that the drivetrain will be protected even if actual motor excitation torques are significantly greater than those predicted.

E. Coupling Designs

Various configurations of the rubber damper coupling are used to suit the requirements of the application. In particular, for turbomachinery applications, a lighter weight (and lower cost) coupling is desired for the driven machine input shaft. In this case a hybrid coupling similar to those of Figures 10.18 and 10.19 is used.

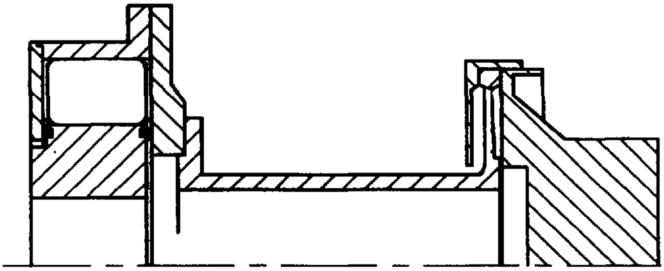


FIGURE 10.18 Hybrid diaphragm.

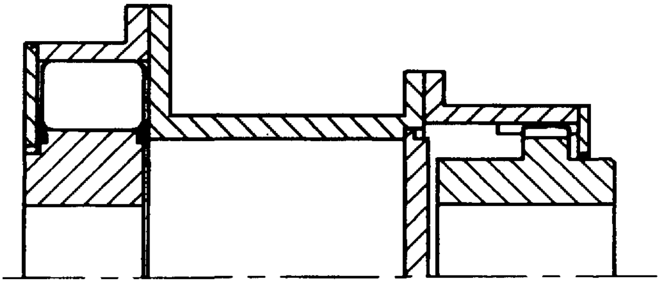


FIGURE 10.19 Hybrid gear.

It must be realized that a damper coupling will be larger and heavier than a nondamping coupling such as a gear-type coupling. Generally it will be about 50% greater in diameter and about twice as heavy as a gear coupling. This additional size and weight can be important with respect to connected machine bearings and must be taken into account at the initial design stage.

III. RETROFITTING NONLUBRICATED COUPLINGS

A. Introduction

Nonlubricated couplings have gained wide acceptance in the petrochemical and process industries. With most new turbomachinery packages being supplied with nonlubricated (dry) couplings, there is an increasing interest in replacing many of the older gear couplings in existence with nonlubricated versions. Properly applied nonlubricated couplings have improved the performance of countless trains initially installed with gear couplings, but improperly applied nonlubricated couplings have created new problems with the drivetrain. This section provides some general guidelines that should be followed whenever a changeover from gear coupling to a nonlubricated coupling is under consideration.

B. Why Change?

Most often interest in replacing a gear coupling with a nonlubricated one occurs when the life of an older gear coupling is reaching its end or when a substantial system upgrade is planned. When the removal of the old coupling is planned in these situations, the opportunity presents itself to take advantage of the benefits associated with nonlubricated couplings.

The first question that arises is, Should the coupling design be changed at all? If the older gear coupling has provided satisfactory service over many years, it may be best to "stick with what works." The cost of changing the design is more than just the cost of the coupling. It also includes the cost of possible changes to the lube system, the coupling guard, and, if your maintenance mechanics are not used to the requirements of a nonlubricated coupling, personnel training.

If the cons of the gear coupling outweigh the pros of a nonlubricated coupling and you have decided to make the leap, nonlubricated couplings can provide several big advantages, including no lubrication, lower unbalance forces, lower and more predictable bearing loads (because of fewer misalignments and axial movements of the equipment), and long service life.

C. Proper Coupling Selection—Whose Responsibility?

When the machinery was originally purchased, the original equipment manufacturer (OEM) had the responsibility of reviewing the coupling design. The OEM provided the application information to the coupling manufacturer for proper selection and design. The coupling characteristics were then given to the OEM, who determined whether the effects on the train would be acceptable. Many systems require that the coupling design be refined during this process to "tune" the lateral and torsional responses. Since the OEM is usually not involved with coupling retrofits years after the train began operating, the responsibility of ensuring that the new coupling design will not adversely affect the train shifts to the end user who requested the retrofit.

When specifying the nonlubricated coupling, be certain to provide the coupling manufacturer with sufficient information to obtain the proper fit of the coupling to the equipment. It is critical to resolve any questions regarding hub-to-shaft fit before manufacture of the coupling is started. If the equipment interfaces are shaft ends, specify the coupling bore dimensions and tolerance exactly. Also specify the required hub fit length and tolerance. If you do not have the specifications of the old gear coupling hub, specify the shaft size and required interference rate, measure the shafts with micrometers, and check the slip torque at the hub/shaft interface. If the shafts are tapered, specify the amount of taper and supply the coupling manufacturer with plug gages from a matched ring gage and plug gage set—where you have verified that the ring gage as well as the plug gage provides proper contact to the shaft. If this is not possible, consider sending the coupling manufacturer the hub from a spare coupling. If the equipment interfaces are integrally flanged, be certain to specify the flange thickness tolerance. Specify the rabbet size, tolerance, and length (along with bolt hole size and location tolerances), including countersinks to clear the bolt head radii, any jacking holes, and whether the bolts can be put in from the equipment side. Be certain to specify who is to supply any new bolts and nuts at these connections.

Make certain that the coupling operating environment is adequately defined, including any unusual temperatures or corrosives. Make certain that all accessibility issues have been addressed. Check to see if the coupling flange is in an equipment tunnel. Verify how the coupling guard is assembled (Is it split?, Can it slide back? etc.).

D. Match the Coupling

Some users have in-house experts who perform the train analysis again with the new coupling characteristics; others turn to independent consultants for this work. Since the costs associated with having a new torsional and lateral analysis can be very high, the accepted practice is to "match" the characteristics of the new nonlubricated coupling with those of the gear coupling. If the new nonlubricated coupling characteristics cannot be matched to the old gear coupling, the resulting operation is effectively a "crap shoot," so a complete lateral and torsional analysis is unavoidable, and the resulting costs may outweigh the benefits of retrofitting.

1. Significant Considerations

The most significant characteristics to consider when attempting to match the nonlubricated coupling to the gear coupling are weight, location of the center of gravity, and torsional stiffness. Other characteristics that must be at least comparable are the balance classification and the coupling's first lateral mode of vibration. In addition, there are a few characteristics associated with nonlubricated couplings that must be considered. These include space restrictions (nonlubricated couplings are often larger in O.D.), temperature rises due to windage, axial natural frequencies, axial and angular capacities, and handling and/or installation techniques.

a. Weight and Half-Coupling Effective Center of Gravity (C.G.) Locations. The effective center of gravity is one of the most critical characteristics to consider, especially for machines with a high sensitivity to unbalance, e.g., a machine whose operating speed is close to its critical speed, such as a high-speed compressor. The sensitivity of a rotor to unbalance is a function of the coupling's half-weight (that portion of the coupling overall weight acting on the rotor of interest) and the distance of this half-weight from the rotor's bearing centerline. Increasing the half-weight or its distance from the bearing can adversely affect the rotor's sensitivity to unbalance and/or its lateral critical speed.

The way in which the half-coupling's effective c.g. is calculated is to assume that negligible moment from the coupling weight can be transmitted across the flex point, thereby treating the coupling's flex point (disc pack, flexible diaphragm, or gear mesh) as a pin joint. The closer the flex point is to the equipment bearing, the smaller the resulting moment from coupling weight causing the shaft to deflect.

The rotor's lateral deflections are a function of this c.g. distance cubed, while the rotor's critical speed is a function of this distance squared. In analyzing the influence of various couplings on a rotor's vibration response, good results have been found by considering the product of the coupling's half-weight times this distance to the first power. This method is referred to in API 617. This coupling characteristic is generally referred to as the coupling's "overhung moment." (See Figure 10.20.)

When it is not possible to duplicate both the weight and the c.g. location, matching this overhung moment has been found to be a suitable criterion for successful operation. Successful operation usually results when the retrofitted coupling is found to match the replaced coupling within the guidelines listed in Figure 10.21.

Note: When the coupling weight is a significant portion of the

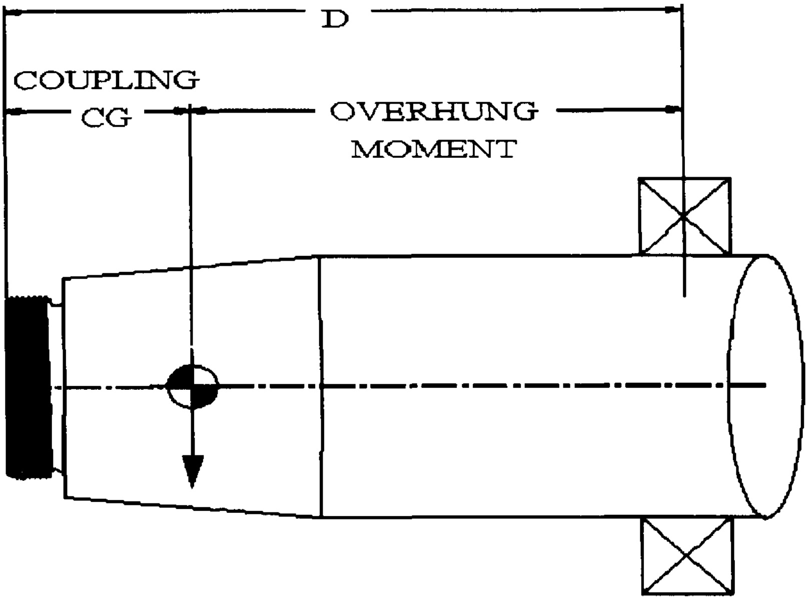


FIGURE 10.20 Moment equivalent.

For operating speeds less than 3600 rpm- overhung moment matched within 20%.

For operating speeds between 3600 and 6000 rpm - overhung moment matched within 15%.

For operating speeds greater than 6000 rpm - overhung moment matched within 10%.

FIGURE 10.21 Weight and center-of-gravity match guidelines.

rotating weight, or if the equipment has been found to be sensitive to vibration, every effort should be made to match or reduce the nonlubricated coupling's overhung moment to that of the existing coupling.

b. Torsional Stiffness. Often couplings are used to tune the torsional resonant frequencies of multibody trains. Since most applications can be effectively represented in lumped mass form with the couplings treated as torsional springs and the equipment rotors as lumped inertias, close attention must be paid to the coupling's torsional stiffness whenever a machine is being retrofitled to a nonlubricated coupling.

In general, the gear mesh flex point of the existing gear coupling will be torsionally stiffer than the flex point of the replacement nonlubricated coupling. Consequently, the design of the nonlubricated coupling will usually have to be tailored to the application if it is to match the gear coupling.

The most common coupling design activity is adjusting the wall of the coupling's spacer (tubular piece usually found between the coupling's flex points). Often, the activity of stiffening parts of the nonlubricated coupling to match the stiffness of the gear coupling leads to an increase in weight of the parts being stiffened, causing the coupling designer to look elsewhere in the nonlubricated coupling to reduce the weight of parts that contribute less to the coupling's torsional stiffness. In the case where the coupling is relatively long in comparison to its diameter, a successful match can usually be obtained. However, for shorter couplings, alternative materials may have to be used or the end result may be a somewhat heavier coupling, where the overhung moment guidelines described in Figure 10.21 are often employed.

Successful operation usually results when the retrofitted coupling is found to match the replaced coupling within the guidelines listed in Figure 10.22. *Note:* When torsional pulsations are known to exist in the train, every effort should be made to match the nonlubricated coupling's torsional stiffness to that of the existing coupling.

2. Other Characteristics to Consider

a. Balance Classification. Nonlubricated couplings inherently exhibit better balance repeatability than gear couplings. Unlike gear cou-

For two-body trains with speeds to 3600 RPM - torsional stiffness matched within +/-25%.

For three-body trains without a gearbox and speeds less than 6000 RPM - torsional stiffness matched within +/-20%.

For all other trains - torsional stiffness matched within +/-15%.

FIGURE 10.22 Torsional stiffness match guidelines.

plings, which require a certain amount of clearance for the gear mesh to slip and tilt, nonlubricated couplings accommodate equipment movements through material flexure and therefore do not require any clearances to operate successfully. Generally, it is recommended to specify that the nonlubricated coupling meet or exceed the balance requirements of the gear coupling being replaced.

b. Lateral Coupling Modes. The first lateral mode of the coupling is generally calculated as a tube supported on each end at the flex points. This *simply supported* tube is able to bend and whirl in this defective *bowed* shape. Consequently, most coupling arrangement drawings will define a simply supported tube lateral critical speed (LCS) (see Figure 10.23).

It is important to note that this "coupling" LCS mode is for the center section of the coupling only. It is in no way an indication of the system's lateral critical speed. The coupling's simply supported tube LCS also generally does not reflect any contributions or deflections of the equipment and associated mounting hubs. The coupling's effect on the equipment's lateral critical speeds are more a function of the overhung moments, so the usefulness of the "coupling" LCS for simple supports is limited to ensuring that the coupling's "whip" mode is sufficiently far from the operating speed so as not to influence the equipment's LCS modes (see Figures 10.24A and 10.24B) or whirl around on its own. Therefore, it is generally recommended that the coupling's first lateral mode be at least 50% above operating speed. In the case of very long equipment separations at high operating speeds this may not be possible, in which case a more in-depth evaluation should be undertaken.

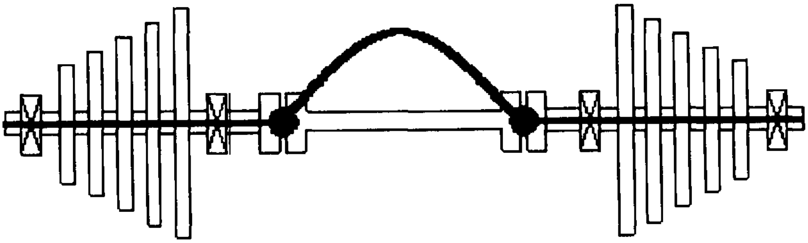


FIGURE 10.23 LCS mode of coupling.

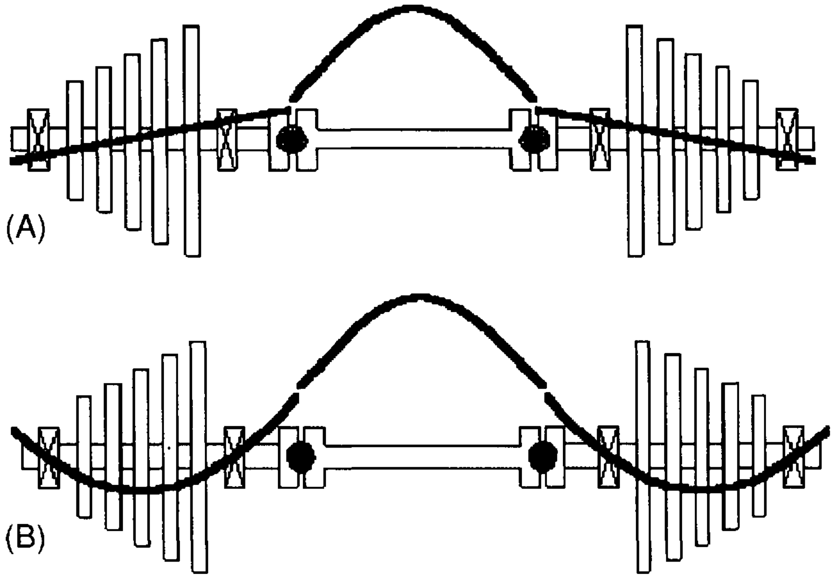


Figure 10.24 Typical system lateral modes.

c. Space Restrictions. For a given torque capacity, nonlubricated couplings are often somewhat larger in diameter than their gear coupling counterparts. Therefore, it is a good idea to provide the coupling manufacturer with a drawing of the existing coupling guard along with any other descriptive information regarding housing restrictions, piping, etc. to ensure that the coupling will physically fit in the application and that if there are any modifications required to the equipment they can be properly planned for prior to the coupling installation.

d. Temperature Rises and Windage Effects. When couplings operate at high speeds in oiltight enclosures, they shear the air; the maximum air velocity is found at the coupling surface, and the minimum air velocity is found at the enclosure surface. The friction between adjacent air layers generates heat. Since nonlubricated couplings are often larger in diameter than gear couplings, they often generate more heat due to these windage effects. The amount of heat that will be generated is a function of the coupling's operating speed and the geometry of the coupling and its surrounding enclosure.

The closer the rotating coupling is to the surrounding nonrotating parts (housing and guard), the more heat will be generated. As a general *rule of thumb,* when the coupling surface (O.D. and faces) is less than 2 in. from the nonrotating components (coupling guards and housings), calculations should be performed to ensure that the resulting temperatures will be within safe limits. The generation of turbulence can usually be minimized by incorporating shrouds in the coupling design in the areas of bolt heads and nuts. If the appropriate information about the surroundings is supplied, most coupling manufacturers can provide estimates for horsepower losses and heat generation.

If this analysis indicates a problem, modifications to the housing, coupling, or coupling guard may be required. It is worth pointing out that if the nonlubricated coupling is being retrofitted to replace a continuously lubricated gear coupling, the guard temperature can be significantly lowered by continuing to use the oil spray system.

Another potential problem that can be encountered when retrofitting nonlubricated couplings in oiltight enclosures is related to pressure differentials. Although pressure problems are not usually related to heat generation, they can cause a different type of problem. Negative pressures can be created in the area of the equipment seals where oil may be sucked from the machines into the coupling enclosure. This situation is usually found only when large disc-shaped parts of the coupling are mounted on the equipment shafts near the seals (usually associated with the "*reduced moment*" variety of coupling). The disc rotation creates an air flow from the slower moving I.D. to the faster moving O.D., resulting in pressure differentials that can cause a loss of oil in the machine housings.

Typical solutions to such negative pressure problems often involve installing a baffle plate (see Figure 10.25) into the guard or housing or in some cases installing an air breather (see Figure 10.26) tube in front of the seal to increase the pressure at that location.

The topics of windage and the proper use of baffles, vents, and drains are covered in greater detail in Section 8.I.B.

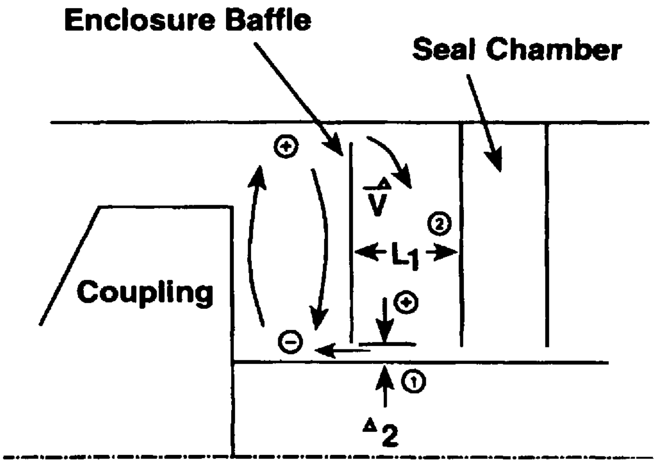


FIGURE 10.25 Example of baffle plate.

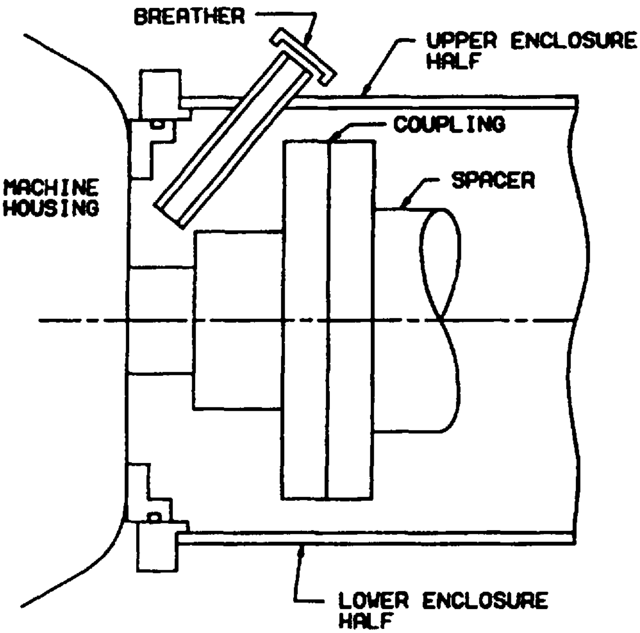


FIGURE 10.26 Example of air breather tube.

e. Axial Natural Frequencies. Since the flex points in nonlubricated couplings are metallic springs, nonlubricated couplings exhibit a potential mode of vibration called the axial natural frequency (A.N.F.) that is not found in gear couplings. This A.N.F. results because the mass of the center member (between flex points) is supported between two axial springs (the two flexible elements typically found on each side of the coupling) (see Figure 10.27). Thus nonlubricated couplings differ from gear couplings in their potential to vibrate axially (along their longitudinal axis). However, this mode is unlike lateral and torsional modes because it is difficult to excite and is isolated to the coupling itself. Potential excitation factors include out-of-square faces and thrust bearing runout.

Because most nonlubricated couplings either exhibit nonlinear axial spring rates or are well damped, large axial deflections are not found at or near their A.N.F.'s. However, when retrofitting a gear coupling to single-element convoluted styles (having linear stiffness) that exhibit an undamped response to external forced vibration, it is good practice to attempt to design the nonlubricated couplings to avoid having calculated A.N.F.'s within 10% of a running speed or 10% of two times running speed. These running speeds are generally considered not to include transient or start-up speeds. The coupling design activities generally will include adjusting the weight of the center member and occasionally adjusting the stiffness of the flex elements.

f. Axial and Angular Capacities. Most machines require flexible couplings to accommodate movements of the machine during opera-

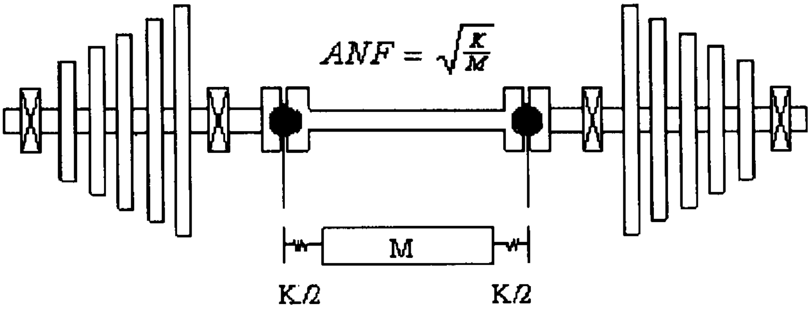


FIGURE 10.27 Coupling axial natural frequency model.

tion. Although the machines can be aligned very accurately, this alignment will not be maintained in operation. The most common cause of these machinery movements is differences in thermal growth of shafting, casings, and support structures (see Figure 10.28). Other commonly encountered contributing factors include piping forces (either temperature or pressure-related) and changes in properties of fluids being handled (see Figure 10.29). Since the equipment will move, it is the coupling's job to accommodate these movements with as little effect on the equipment as possible.

The equipment will move in three basic directions: vertically (up and down), horizontally (sideways), and axially (along the shaft axis). Since these movements will not be the same on each side of the equipment, the machinery shaft centerlines will also become skewed relative to their original axis. To accommodate the offset that often results between the equipment centerlines, flexible couplings are used that have a flex element on each end of the coupling.

To the flexible coupling, all of these combinations of equipment movements boil down to two basic parameters: axial displacement and angular misalignment. This is because all lateral machinery movements are resolved as angles at the coupling's flex elements. Hence the coupling's capability to handle these two parameters is usually called out

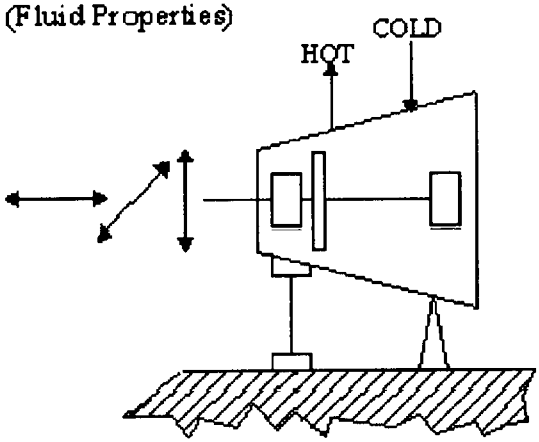


FIGURE 10.28 Typical equipment movement from fluids.

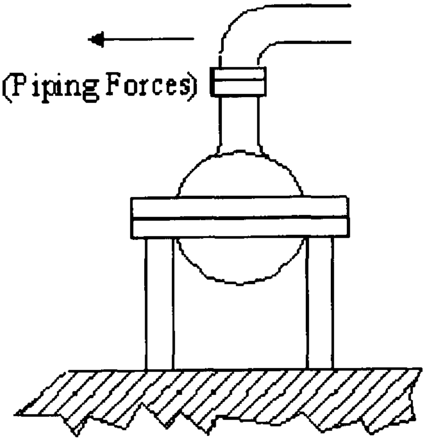


FIGURE 10.29 Typical equipment movement from pipe forces.

by the coupling manufacturer on most coupling arrangement drawings as *axial displacement capacity* and *angular misalignment capacity*. Since the operating angles seen at each end of the coupling are usually not equal, it is customary to give the angular capacity per *end* while the axial capacity is typically per *coupling*.

Gear couplings are designed to handle these misalignments very differently from nonlubricated couplings. Gear couplings are effectively crowned splines with a built-in small amount of radial clearance and backlash between mating male and female teeth. These small clearances are necessary to allow the male teeth to slide and tilt along the mating female teeth. It is this sliding and tilting that requires that the gear coupling be lubricated. Most gear couplings are designed to operate continuously at 1/4° and are very generous in their ability to absorb axial displacement of the equipment.

The flexible elements of nonlubricated couplings deflect by allowing the material to bend and stretch within specified limits. Because the material flexes instead of sliding, there is no need for lubrication. Flexible elements must be strong enough to transmit the required power but thin enough to deflect without failing or causing unacceptable loads on the equipment bearings. Like gear couplings, most nonlubricated couplings are also designed for 1/4°, but *the nonlubricated coupling's abil-*

ity to absorb axial movements of the equipment is usually less than that of a gear coupling. For this reason, the axial movements of the machines must be known before the nonlubricated coupling can be selected. (Be certain to specify whether or not you have accounted for the coupling's thermal growth whenever you specify the axial thermal growth of the machinery.)

If nonlubricated couplings are operated within their specified capacities, they will far outlast their gear coupling counterparts and will impose more predictable and significantly lower loads on the machines. However, if they are exposed to conditions that exceed their stated capacities, the results can be extremely detrimental, including the possibility of catastrophic failure. For these reasons, it is important to pay close attention to the axial and angular capacities to make certain that the retrofitted nonlubricated coupling is operated within its intended useful range.

g. Handling and Installation Techniques. The most important activity in the installation of a nonlubricated coupling is an adequate review of the installation procedure in advance, especially if the mechanics are used to installing only gear couplings. Unlike gear couplings, installations of nonlubricated couplings must involve precise measurement of the distance between the coupled equipment. Nonlubricated coupling manufacturers provide shims to adjust for slight differences in equipment spacing and coupling length from the nominal values, but this adjustment is usually between 1/8 in. total, or ± 1/16 in., to 1/4 in. total, or ± 1/8 in.

Most nonlubricated couplings are designed to be installed so that they run in a more *relaxed* state during steady-state conditions. Since machine shafts typically grow toward each other, most nonlubricated couplings are intended to be "stretched" in the cold (installed) condition, so the coupling manufacturer intentionally designs the nonlubricated coupling to be "short" by a specific amount. Usually this amount is equal to or just less than the expected axial movements of the equipment during normal operation.

Another difference between nonlubricated couplings and gear couplings is the way in which rabbets (or spigots) are to be cleared when the center member is being installed. With gear couplings, you simply slide the sleeve (internal gear) over the hub (external gear) to clear the rabbets. With nonlubricated couplings, the restoring force tends to "fight back," so you have to mechanically collapse the flexible element axially to hold it in place while the center section is brought in. Since these flex elements are sensitive to axial movement, you must be careful not to overcollapse them past their stated capacity (usually half of the coupling's capacity on the drawing). Many coupling manufacturers provide separate hardware for the sole purpose of properly collapsing the flex elements. Since this activity is not found with gear coupling installations, it is critical to make certain that everyone involved knows that this collapsing hardware is not part of the coupling and must be removed (and double checked for removal) prior to start-up.

The flexible elements of nonlubricated couplings are more sensitive to abusive handling, so the installers must take precautions not to scratch, dent, or ding these areas during installation.

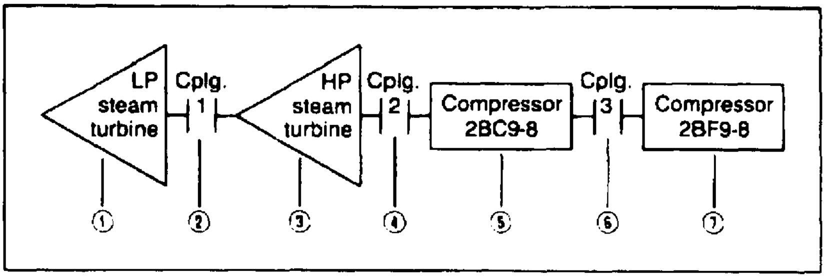
It is important to have all coupling and equipment drawings, instruction sheets, tools, lifting devices, and parts on hand prior to scheduling the actual installation. It is recommended that the phone number of a contact at the coupling manufacturer be available for any unforeseen circumstances.

Since it is recommended that the equipment alignment be rechecked after the coupling is installed, it is important to know what surfaces of the nonlubricated coupling can be used for indication purposes.

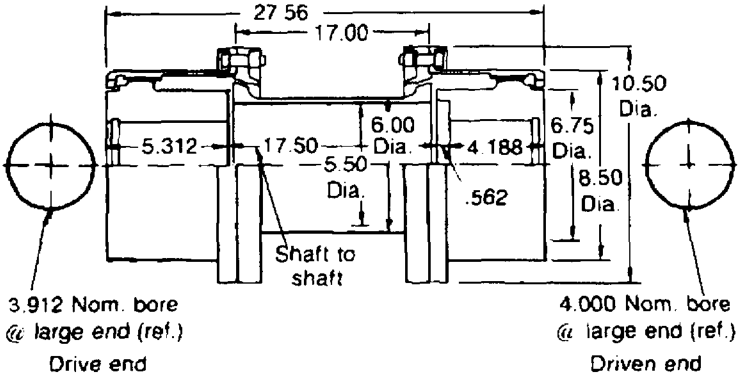
Nonlubricated couplings sometimes have more fasteners than gear couplings. It is important to install the bolts in the direction shown on the coupling drawing and to use proper torque wrenches while tightening them to the prescribed value. The hardware is generally supplied in weight-balanced sets and must not be mixed with other sets. Use only the coupling manufacturer's bolts and nuts. Pay attention to the matchmarks and serial numbers on the coupling to ensure that each part is properly positioned in the assembly. If any questions arise, do not hesitate to call the coupling manufacturer for clarification.

EXAMPLE OF RETROFITTING

We will look at replacement of one coupling in a syngas train (Example Figure 1). We will be looking at coupling No. 2 in position 4.



Example Figure 1 Syngas train.

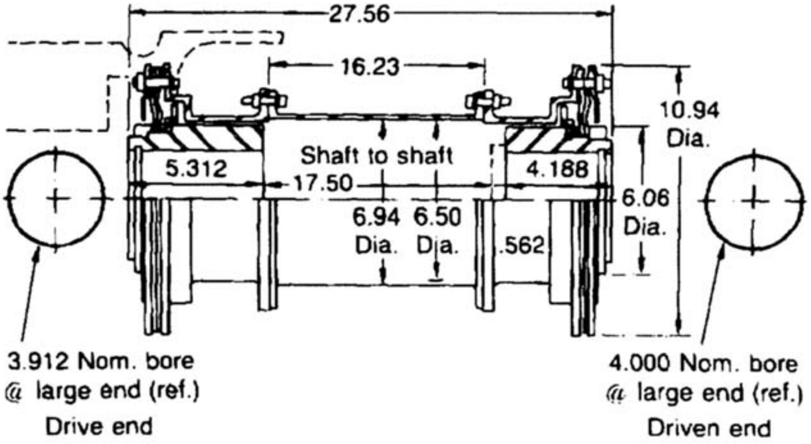


Example Figure 2 Reduced moment gear coupling (position 4). (Courtesy of Ameridrives International.)

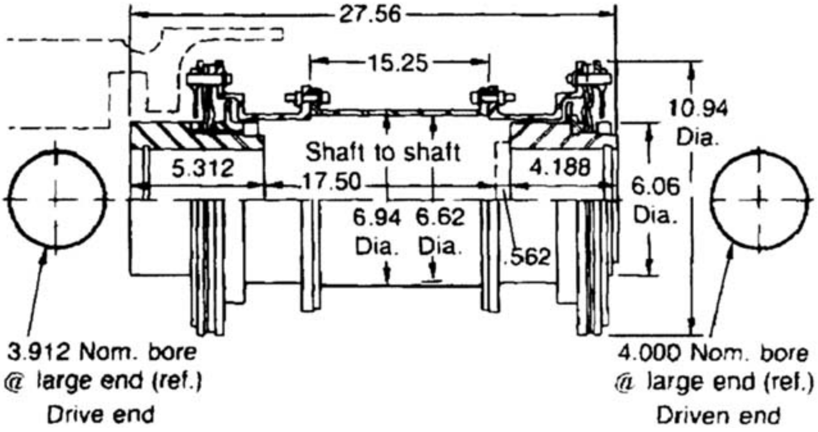
The combined horsepower requirements for coupling 2 were 27,500 hp at a speed of 10,300–11,000 rpm (see Example Figure 2).

When an initial analysis was done, the coupling shown in Example Figure 3 was first proposed, but after a review of the equipment layout it was found that the proposed diaphragm coupling would interfere with the turbine housing.

After obtaining the housing layouts it was decided that the diaphragm pack had to be moved forward on the turbine shaft (see Example Figure 4). However, if it were moved too far it could upset the system critical. It was moved forward only enough to provide clearance so that during operation the coupling bolts would not contact the hous-



Example Figure 3 First proposed diaphragm coupling (interference with turbine housing). (Courtesy of Ameridrives International.)



Example Figure 4 Final design of diaphragm coupling (semireduced moment-O.D. bolts with helicoils). (Courtesy of Ameridrives International.)

Characteristics	Original gear cplg	Retrofitted Diaphragm cplg	Retrofitted Dia. cplg
OD (inches)	8.2	10.94	8.97
Comp. weight @ CG both ends	34.3 lb @ 2.74 in.	55 lb @ 2.74 in.	31 lb @ 2.97 in
Torsional stiff.	5.4 x 10^6 in.-lb/rad	6.7 x 10^6 in.-lb/rad	5.4 x 10^6 in.-lb/rad

Example Figure 5 Comparison table of example of mass elastic properties.

ing (approximately 1/4 in. clearance) Calculations had to be made to determine the heat generated from the nuts against this housing, and it was determined that they could generate enough horsepower loss to cause overheating of the coupling guard. Therefore a stepped guard on the coupling and helicoils instead of nuts were used to reduce the horsepower loss and keep the guard temperature within the acceptable range.

Example Figure 5 shows a comparison of the mass elastic properties for the gear coupling, the first coupling proposed, and the final coupling installed in the application. The final coupling was installed in 1986, and no known operating problems have been reported. Initially the vibration levels were reported to be 25% below those of the gear coupling. The guard temperature was measured to be approximately 135°F, which is very close to the calculated value of approximately 140°F.

11

Coupling Nomenclature

Accessory set A package of hardware used to assembly coupling components; may include, fasteners, lube plugs, shims, plate, gaskets, etc.

Adapter plate An auxiliary device required to rigidly hold in alignment the floating sleeve at the drive end of a flex-hub coupling to allow solo operation of the driver without the need to dismount the coupling hub. Also known as a solo plate.

Aligned shafts When aligned, two shafts are coaxial with a specified separation.

Aligning surface The surface used to establish the axis from which a part (or coupling) is located for the purpose of balancing.

Allowable speed Speed related to application conditions and experience that includes consideration of misalignment, loading, and system characteristics. Allowable speed shall not exceed rated maximum speed rating of the coupling.

Angular misalignment, double-engagement The minor angle between the axial centerline of the structure joining the two flexing elements and either of the shaft centerlines. Thus, there are two misalignment angles associated with a double-engagement coupling. These two angles are not necessarily equal, since they are the result of common parallel offset combined with the angular misalignment of the respective shaft end.

Angular misalignment, general The minor angle between two shafts when the two shafts are not coaxial and their axes are not parallel.

Angular misalignment, single-engagement The minor angle between the two shaft centerlines.

Anti sludge gear coupling design A gear coupling, design that minimizes sludge-gathering pockets in continuously lubricated couplings.

Application factor A factor applied to the normal operating torque to account for unusual conditions that may occur unexpectedly such as compressor fouling, or to account for real operating conditions. *See also* Service Factor.

Assembly balance A procedure in which a completely assembled coupling is balanced as a unit.

Assembly check balance A procedure in which an assembly is placed on a balance machine and the potential unbalance is measured. This can be done to a component-balanced coupling or to an assembly-balanced coupling.

Axial displacement (End float) A change in the gap between the shaft ends of two coupled machines, usually caused by thermal expansion.

Axial natural frequency (A.N.F.) A cyclic response frequency determined by the mass of the flexible center element acting against the axial spring rate of the flexible elements. The spring rate may vary with the deflection of the flexible element; therefore, a frequency band corresponding to deflection amplitudes from zero to the maximum allowable amplitude must be considered.

Axial reaction force *See* Transmitted axial force.

Axial reference point The axial position on the shaft of the driving or driven machine (normally the extreme end of the shaft) from which axial distances are measured.

Axial stiffness A measure of a coupling's resistance to axial displacement.

B_{10} life For a universal joint bearing, the life expectancy for a 90% probability of survival.

Backlash The circumferential clearance in the flexible element(s) that is necessary to provide angular misalignment capability and assembly.

Batch-lube gear coupling A coupling designed to be lubricated by a periodically changed charge of grease or oil.

Bending stiffness (Angular stiffness) The ratio of restoring moment to angular deflection. Couplings that slide while accommodating angular misalignment do not possess angular stiffness, as material deflection does not occur.

Bushing An adapter or adapter assembly used to mount the coupling components or coupling assembly on the mandrel.

Center of gravity For a half-coupling, the location at which the weight of that half-coupling can be considered to be concentrated. The half-coupling's effective center of gravity is referenced from the equipment shaft end, with a positive location being beyond the shaft end and a negative location being within the shaft.

Chain coupling A coupling that consists of two hubs with sprocket teeth that engage or mesh with a strand of chain.

Check balance A procedure wherein an assembled coupling is placed on a balance machine and the potential unbalance is measured. This can be done to a component-balanced coupling or to an assembly-balanced coupling.

Clearance-fit bore A bore with a hub bore diameter equal to or larger than the shaft diameter.

Component balance A procedure for improving coupling balance in which the components or factory assemblies are balanced before the coupling is assembled.

Continuously-lubricated gear coupling A coupling designed to be lubricated by a continuous external supply of oil directed at or into the gear mesh.

Continuous torque rating (T_c) The coupling manufacturer's value of torque capabilities of the coupling expressed as a combination of speed, angular misalignment, and axial displacement.

Coupled unbalance That condition of unbalance for which the central principal axis of inertia intersects the shaft axis at the center of gravity.

Coupling mass simulator An auxiliary device designed to correctly simulate both the effective mass and the effective location of the center of mass of a half-coupling with respect to the shaft on which it is mounted.

Coupling rated speed The maximum speed at which the coupling is capable of transmitting the coupling continuous rated torque while being simultaneously subjected to the rated angular misalignment and the coupling rated axial displacement.

Coupling rating The coupling's torque capacity at rated misalignment, axial displacement, and speed.

Coupling flywheel effect (Polar weight moment of inertia) A measure of the potential of the coupling to resist change in speed equal to the product of the weight of the coupling times the square of the radius of gyration (WR^2). The radius of gyration is the radius at which the mass of the part (coupling) can be considered concentrated.

Critical damping coefficient *See* Damping ratio.

Crown diameter The major diameter of the external gear teeth.

Damping The absorption or dissipation of oscillatory energy. Damping may be necessary to limit the buildup of transient or steady-state resonant oscillations in a system.

Damping coefficient The ratio of damping energy during one cycle to the elastic deformation energy.

Damping ratio (Factor of critical damping) The ratio of the actual damping coefficient (C) to the critical damping coefficient (C_c). The critical damping coefficient is a measure of the minimum damping that will allow a displaced system to return to its initial position without oscillation.

Design factor of safety (D.F.S) The factor of safety at the catalog rated conditions of torque, speed, misalignment, and axial displacement. It is used by the manufacturer to establish the coupling rating.

Diametral crown clearance (Tip clearance) The clearance between the outside diameter of the coupling's external teeth (whether on a hub or on a marine spool piece) and the root diameter of the internal teeth in the sleeve when the coupling is in perfect alignment.

Diaphragm coupling (Metallic diaphragm coupling) A coupling consisting of one or more metallic flexible elements in the form of thin circular plates that are attached to one part of the coupling at their outer diameter and to the other part at their inner diameter.

Disc coupling (Metallic disc coupling) A coupling consisting of one or more metallic flexible elements that are alternately attached to the two parts of the coupling, the attachment points being substantially at the same distance from the centerline.

Distance between shaft ends (BSE) The distance from the face of one shaft (including any thread end) to the face of the next shaft. For integral flanges, this measurement shall be from the mating face.

Double-engagement coupling (Double flex coupling) A coupling with two flexing planes. This type of coupling can accommodate parallel offset, angular misalignment, and axial displacement.

Dynamic unbalance That condition in which the central principal axis of inertia is not parallel to and does not intersect the shaft axis.

Elastomeric compression coupling A coupling that transmits torque through one or several elastomeric elements, often in the form of blocks, wedges, or a single insert.

Elastomeric flexible element A coupling that transmits torque through one or several elastomeric elements. Can be categorized into two general types, compression and shear, depending on the way torque is transmitted through the flexible element.

Elastomeric shear coupling A coupling that transmits torque through an elastomeric element that is principally loaded in shear. The flexible element that is placed into compression between axially extending lugs or pockets of two hubs. A single elastomeric element is usually able to accommodate angular misalignment, parallel offset, and axial displacement.

Electrically insulated coupling A coupling that prevents the flow of electric current from one shaft to the other through the coupling by inserting insulation between the flanges and around the flange bolts.

End float *See* Axial displacement.

Endurance limit The fatigue limit of a coupling component subjected to combined constant and alternating stresses. Beyond this limit the material can be expected to fail after some finite number of cyclic loads. Below this limit the material can be expected to have infinite life (or a factor of safety greater than 1).

Factor of safety (F.S.) A factor that is used to cover uncertainties in a coupling design: analytical assumptions in stress analysis, material unknowns, manufacturing tolerances, etc. Under given design conditions the F.S. is the ratio of strength (or stress capacity) to actual predicted stress, where the stress is a function of torque, speed, misalignment, and axial displacement.

Fatigue factor of safety The factor of safety at the published rated conditions of torque, speed, misalignment, and axial displacement. It is used by the manufacturer to establish the coupling rating.

Flex hub coupling A coupling that has the external gear teeth on the hubs and internal teeth in the sleeves.

Flexible element The part of a coupling that provides flexibility.

Flexing length The axial distance between the effective flexing planes of a double-engagement coupling.

Floating shaft Typically, a removable center member of a flexible coupling assembly that is composed of two standard half-couplings (single-engagement or single flexing) mounted on either end of a piece of shafting (solid or tubular) and is used to connect the driving and driven shafts of equipment, which are separated axially by greater than normal distances.

Flooded-mesh coupling A continuously lubricated coupling in which the gear meshes are completely submerged in oil during normal operation.

Gear coupling A coupling that transmits torque and accommodates angular misalignment, parallel offset, and axial displacement by a relative rocking and sliding motion between mating, profiled gear teeth.

General-purpose coupling A coupling that transmits power between the rotating shafts of two machines for general-purpose applications. Such applications will typically require couplings to transmit power at speeds not exceeding 3800 rpm between machines in which the first lateral critical speed is above the running speed range (stiff shaft machines). This coupling is often a standard type that can be selected from a catalogue.

Grid coupling A coupling that consists of two flanged hubs with slots or grooves in the flanges, connected by a serpentine mechanical flexible element.

Hardware The nuts, bolts, washers, etc. that are used to attach the various coupling components together.

Half-coupling The composite of all the components of the coupling attached to and supported from one shaft; includes an appropriate portion of the spacer assembly in the case of a double-engagement coupling or of the flexing elements of a single-engagement coupling.

High cycle fatigue The regime characterized by low overstress (exceeding the fatigue limit) where life is usually above 10^3–10^5 cycles.

High-speed coupling A coupling designed to operate at a surface velocity above 3200 in./sec. For a high-speed coupling, $V = D \times \text{rpm}/19$, where D is the maximum outside diameter of the coupling in inches.

Horsepower The unit of power that has been adopted for engineering work in the English system. One horsepower (1 hp) is equal to 33,000 ft-lb (of work) per minute, or 550 ft-lb/sec.

Hub The part of a flexible coupling that is machined for mounting on a shaft.

Hub bore A cylindrical or conical hole in the hub of a coupling whose axis is coincident with the rotational axis of the coupling.

Intermediate-speed coupling A coupling designed to operate at a surface velocity of 2300–3200 in./sec. For an intermediate-speed coupling, $V = D \times \text{rpm}/19$, where D is the maximum outside diameter of the coupling in inches.

Interference fit A condition where the hub bore diameter is smaller than the shaft diameter.

Key A metallic mating load-transmitting member placed in a groove in both the shaft and hub.

Keyseat (Shaft keyway) The axial groove in the shaft that holds the key in the proper location.

Keyway The axial groove in the hub that holds the key in the proper location.

Lateral offset The lateral distance between the centerlines of two coupled shafts that are not parallel measured at the axial reference point of the driving machine shaft.

Lateral stiffness, radial The ratio of force to displacement while the coupling is being deflected in a plane perpendicular to the coupling axis of rotation.

Limited-end-float coupling A coupling designed to limit the movement of the coupled shaft ends with respect to each other where one shaft has no thrust bearing for centering. A limited-end-float design is commonly used in couplings for sleeve-bearing motors.

Low cycle fatigue The regime characterized by high overstress (exceeding the yield limit) where life is usually below 10^3–10^4 cycles.

Low-speed coupling A coupling designed to operate at a surface velocity below 2300 in./sec. For a low-speed coupling, $V = D \times$ rpm/19, where D is the maximum outside diameter of the coupling in inches.

Machine rated speed The highest rotational speed at which the machine rated torque is required to be transmitted continuously by the coupling.

Machine rated torque (T_m) The maximum continuous torque required to be transmitted by the coupling. Mean torque is the time-average torque over a few revolutions and does not include variations such as those associated with reciprocating machines.

Mandrel (Arbor) A cylindrical part on which the coupling component or assembly is mounted for balancing.

Marine style coupling A coupling whose external gear teeth are on the spacer and whose internal teeth are in the sleeves.

Maximum allowable (rated) speed The highest speed (in revolutions per minute) at which the manufacturer's design will permit continuous operation.

Maximum allowable temperature The maximum continuous temperature for which the manufacturer has designed the equipment.

Maximum bore The largest bore for a specified hub diameter consistent with the torque rating and keyway depth (if any) of the coupling.

Maximum continuous angular misalignment The maximum misalignment (both angular misalignment and parallel or lateral offset) the coupling is able to tolerate for unlimited periods expressed either as a single value at the coupling rated speed when the coupling is transmitting its continuous rated torque and is simultaneously subjected to the coupling rated maximum continuous axial displacement, or as an interrelated function of speed, torque, and axial displacement.

Maximum continuous axial displacement (d_c) The maximum axial displacement the coupling is able to tolerate for unlimited periods expressed either as a single value at the coupling rated speed when the coupling is transmitting its continuous rated torque and is simultaneously subjected to the coupling rated maximum continuous annular misalignment, or as an interrelated function of speed, torque, and angular misalignment.

Maximum continuous rating (MCR) A rating determined by the manufacturer to be the torque capacity at which the coupling can safely run continuously and have an acceptable design factor of safety.

Maximum continuous speed The speed (in revolutions per minute) at least equal to 105% of the highest speed required by any of the specified operating conditions.

Maximum continuous temperature The maximum temperature at which the coupling will continuously transmit the coupling continuous rated torque.

Maximum continuous torque The torque capacity that a coupling can safely run at continuously and that has an acceptable design factor of safety.

Maximum momentary rating (MMR) The rating determined by the manufacturer to be the torque capacity that a coupling can experience without ultimate failure where localized yielding (damage) of one of its components may occur. A coupling can withstand this occurrence for one brief duration. After that, the coupling should be inspected and possibly replaced. (Also sometimes called the Short-circuit torque rating.)

Maximum speed A speed related to centrifugal stresses in the coupling. The manufacturer's definition of the rotating speed limit of the coupling is usually defined in revolutions per minute.

Mechanical flexible element coupling (Mechanical contact coupling) A form of flexible element that accommodates misalignment by the sliding of mating surfaces. Such an element does not have a free-state position. It can be at rest at any combination of axial and angular positions within its flexible capability. Mechanical flexible elements resist change in axial and angular position. The resisting forces are mainly a function of shaft torque and coefficient of friction between the mating surfaces. Examples of mechanical flexible elements are gear, grid, and pin- and-bushing couplings.

Metallic diaphragm coupling A coupling consisting of one or more metallic flexible elements in the form of thin circular plates that are attached to one part of the coupling at their outer diameter and to the other part at their inner diameter.

Metallic disc coupling A coupling consisting of one or more metallic flexible elements that are alternately attached to the two parts of the coupling, the attachment points being at substantially the same distance from the centerline.

Metallic flexible element coupling A form of flexible element that accommodates misalignment by material deformation of a metallic member. These elements are very much like springs in that they have a free-state shape and will resist a change in shape with a reaction force. Examples of metallic flexible elements are disc, convoluted diaphragm, and contoured diaphragms.

Minimum allowable speed The lowest speed (in revolutions per minute) at which the manufacturer's design will permit continuous operation.

Minimum continuous speed The lowest rotational speed at which the coupling is required to operate continuously.

Minimum operating temperature The lowest temperature at which the coupling is required to transmit torque and/or accommodate misalignment or axial displacement.

Misaligned shafts Shafts that are parallel offset, or angularly or axially misaligned or any combination thereof.

Moment simulator An auxiliary device required to simulate the moment of the half-coupling. A moment simulator shall also be designed to serve as a solo plate.

Momentary torque limit A torque limit that corresponds to a minimum factor of safety of 1.0 with respect to the component's material yield strength expressed as a combination of speed, angular misalignment, and axial displacement.

Neutral state of a flexible, element coupling The state in which there is no net axial force imposed on the coupling.

Neutral state of a gear coupling The state when the meshing pairs of gear teeth are axially centrally located with respect to each other, that is, with equal axial clearance in each direction.

Nominal bore A term commonly used to identify the basic bore size without tolerance.

Normal operating point The point at which normal operation is expected and optimum efficiency of the driver or driven equipment is desired. This is usually the point at which the vendor certifies that performance is within the tolerances stated in the standard.

Normal speed (N_{normal}) The speed corresponding to the normal power, stated in revolutions per minute.

Normal torque The torque that the coupling is to transmit during normal operation and usually also during extended operation. The coupling is designed to transmit normal torque indefinitely without damage or degradation.

Parallel offset misalignment The lateral distance between the centerlines of two coupled shafts that are parallel but not in the same straight line.

Peak torque rating The torque rating that corresponds to a minimum factor of safety of 1.15 with respect to the component's material yield strength, expressed as a combination of speed, angular misalignment, and axial displacement.

Pilots (Rabbets) Surfaces that support a coupling component, subassembly, or assembly upon which another coupling component is mounted or located.

Pin-and-bushing coupling A coupling that consists of two flanged hubs with pins on one flanged hub fitting into bushings in the mating flanged hub.

Polar mass moment of inertia The measure of resistance to a change in rotational velocity about an intended rotational axis. It can be derived by dividing the flywheel effect WR^2 by the acceleration due to gravity: $J = WR^2/g$ (lb-ft/sec^2), where R is the Radius of gyration (ft), W is the weight (lb), and $g = 32.2$ ft/sec^2.

Potential unbalance The maximum probable net unbalance of a coupling after installation. Potential unbalance results from a condition of the residual unbalance of individual components and subassemblies and possible eccentricity of the components and subassemblies due to runout and tolerances of the various surfaces and registers. The numerical value of the potential unbalance is the square root of the sum of the squares of all the contributory unbalances. Typical contributory unbalances are (1) the residual unbalance of each component or subassembly, (2) errors in the balance of each component or subassembly resulting from eccentricity in the fixture used to mount the component or subassembly in the balance machine, and (3) the unbalance of each component or subassembly due to eccentricity resulting from clearance or runout of the relevant registers or fits.

Quill shaft coupling A coupling that is both laterally and torsionally flexible. Angular misalignment, parallel offset, and torsional fluctuation are accommodated by elastic deformation of a relatively long slender shaft. Quill shaft couplings cannot accommodate for axial displacement.

Rated speed The maximum speed at which the coupling is capable of transmitting the coupling continuous rated torque while simultaneously being subjected to the rated angular misalignment and the coupling rated axial displacement.

Residual unbalance The level of unbalance remaining in a component or assembly after it has been balanced either to the limit of the capability of the balancing machine or in accordance with the relevant standard.

Rigid hub A hub that mounts on a shaft. Rigid hubs do not incorporate flexible elements.

Rigidifying hardware Hardware used in balancing to make the coupling rigid and aligned with its designed rotating axis.

Root diameter The diameter of the root circle of the internal gear teeth.

Rough bore A centrally located axial hole produced with dimensions and tolerances in accordance with the practice of each manufacturer. It is not appropriate for mounting.

Service factor (S.F.) (Application factor) A factor that accounts for unusual and occasional loading.

Shaft penetration factor The percentage of the shaft length within the confines of the hub that is, for the purpose of calculating torsional rigidity, assumed to be free from restraint at the shaft/hub interface.

Shear elastomeric coupling A coupling that transmiter torque through an elastomeric element that is principally loaded in shear. The element may be in the form of a type, a bellows (with one or more convolutions), or a diaphragm. A single elastomeric element is usually able to accommodate angular misalignment, parallel offset, and axial displacement.

Single-engagement (single flex) coupling A coupling that has only one flexing plane. This type of coupling can accommodate angular misalignment and axial displacement. Certain types of single-engagement couplings can also accommodate offset misalignment.

Sleeve A component with internal teeth.

Sliding block coupling A coupling that consists of two jawed (slotted) hubs engaged with a mating center member. ("Sliding block" is the preferred term for what was formerly called an Oldham coupling.)

Solid hub A hub that has no bore.

Solo plate An auxiliary device required to rigidly hold in alignment the floating sleeve at the drive end of a flex hub coupling to allow solo operation of the driver without the need to dismount the coupling hub. Also known as an adapter plate.

Spline bore A series of parallel keyways formed internal with the bore and mating with the corresponding grooves cut in a shaft. Splined hubs and shafts are used when relatively heavy torques are to be transmitted. The most common splines conform to standards published by SAE and ANSI and are in the form of involutes or straight, parallel sides.

Static unbalance That condition of unbalance in which the central principal axis of inertia is displaced only parallel to the shaft axis.

Steady-state torque (T_s) The torque used to select couplings should be based upon the following equations:

In SI units:

$$T_s = \frac{9549P_{\text{normal}} \times \text{S.F.}}{N_{\text{normal}}}$$

In U.S. units:

$$T_s = \frac{63025P_{\text{normal}} \times \text{S.F.}}{N_{\text{normal}}}$$

where

T_S = steady state torque used to select the coupling (SI, newton-meters; U.S., inch-pounds)
P_{normal} = input power required by the driven machine at the specified normal operating point (SI, kilowatts; U.S., horsepower)
N_{normal} = speed corresponding to the normal power (in revolutions per minute)
S.F. = service factor

Spacer Typically, an integral two-flanged removable center member that connects the single flexible element components of the coupling and connects driving and driven machine shafts while providing a specified axial shaft separation.

Spacer gap length The flange-to-flange dimension between coupling hubs or sleeves in which the coupling spacer is installed. Spacer gap length is not necessarily equal to the distance between the shaft ends.

Special-purpose coupling A flexible coupling designed and manufactured for a specific application. It must operate under special conditions (high torques, angles, axial or speed) or is required to operate continuously for extended periods. Special-purpose couplings are critical to the continued operation of the installation.

Spline bore A bore that has a series of parallel slots formed internal to the bore and mating with corresponding grooves cut in a shaft. The most common spline Bores conform to standards published by other organizations such as SAE, ISO, DIN, and ANSI and are in the form of involutes or have straight parallel sides.

Straight (Finished) bore A concentric axial cylindrical hole finished to tolerances appropriate for mounting.

Swing diameter The maximum diameter of the circular path described by rotating a universal joint.

Tapered bore A concentric axial conical hole finished to tolerances appropriate for mounting.

Torsional critical speed The critical speed of a whole system rather than of a coupling alone. A torsional critical speed exists when the rotating bodies vibrate back and forth relative to each other in the torsional (twisting) direction. The coupling is just the "middleman" in this situation.

Torsional damping The absorption or dissipation of oscillatory rotational energy. Torsional damping may be necessary to limit the buildup of transient or steady-state torsional resonant oscillations in a system.

Torsional natural frequency Any natural cyclic rotational oscillatory rate of a system composed of rotating mass inertias acting in combination with the restraining torsional rigidities of the connected shafts and couplings.

Torsional stiffness The torque required to produce unit angular displacement, either of the whole coupling or of part of the coupling such as the spacer. *Note:* With some types of couplings, the torsional stiffness may not be constant but may be a function of the magnitude of the torque and, with oscillating torques, also the frequency.

Torsional tuning The shifting of one or more torsional natural frequencies of a coupled system to avoid system resonance at a known excitation frequency. Torsional tuning is normally accomplished by varying the torsional stiffness of the coupling.

Torque (Moment of a force) A measure of the tendency of a force to rotate the body upon which it acts about an axis. The magnitude of the moment due to a force acting in a plane perpendicular to some axis is obtained by multiplying the force by the perpendicular distance from the axis to the line action to the force. Torque is commonly expressed in pound-feet or pound-inches.

Total indicator reading (TIR) The difference between the maximum and minimum readings of a dial indicator, or similar device, monitoring a face or cylindrical surface during one complete revolution of the monitored surface. *Note:* For a perfectly cylindrical surface, the indicator reading implies an eccentricity equal to half the reading. For a perfectly flat face, the indicator reading implies an out-of-squareness equal to the reading. If the surface in question is not perfectly cylindrical or flat, the interpretation of the meaning of TIR is more complex and may represent an out-of-round condition.

Transmitted axial force The axial force transmitted through the coupling from one shaft to the other. It is a function of the resistance to deflection of the flexible element or the sliding of the gear teeth.

Transient torque The torque that the coupling is to transmit during unusual or short-term operation. The coupling should be designed to transmit the transient torque for 1000–10,000 cycles without damage or degradation.

Trip speed The rotational speed of the coupling corresponding to the speed at which the independent emergency overspend device operates to shut down a variable-speed prime mover. Where the term is used in relation to a machine train driven by a fixed-speed electric motor, the trip speed shall be assumed to be the coupling speed that corresponds to the synchronous speed at the maximum supply frequency.

WR^2 (Flywheel effect) A measure of the potential of the coupling to resist change in speed. WR^2 is the product of the weight of the coupling times the square of the radius of gyration. The radius of gyration is the radius at which the mass of the part can be considered concentrated. W is the part weight in pounds; R is the radius of gyration in feet or inches; so WR^2 is expressed in lb-ft^2 or lb-in^2. For a cylinder (such as a coupling hub), $WR^2 = W\ (R_o{}^2 - R_i{}^2)$, where R_o is the outside radius and R_i is the inside radius.

Yield limit (Y. L.) A value determined by the manufacturer to be the failure strength limit of a coupling component that will cause detrimental damage. If this limit is exceeded, the coupling should be replaced.

Appendix A

Coupling Manufacturers

	Mechanical element				Elastomeric element			Metallic element			
Manufacturer	Gear	Grid	Chain	U-Joint	Shear	Compression	Combined	Disc	Diaphragm	Misc.	Miniature
Acme, Inc. 825 Main St. Holyoke, MA 01040			×								
Acushnet Co. 744 Belleville New Bedford, MA 02742					×						
Addax P.O. Box 81467 Lincoln, NE 68501								×			
All Power Transmission 3146 Market St. Green Bay, WI 54304				×							
American Vulkan Corporation 2525 Dundee Rd. P.O. Drawer 673 Winter Haven, FL 33882					×						
Ameridrives International 1802 Pittsburgh Ave. P.O. Box 4000 Erie, PA 16514	×			×					×		
Atra-Flex 110 E. Garry Ave. Santa Ana, CA 92707						×					

Boston Gear Division, IMO Industries, Inc. 13 Hayward St. Quincy, MA 02171				×	×	×					
Browning Mfg./Division of Emerson Electric Co. 1935 Browning Dr. Maysville, KY 41056			×		×						
Cline Co., the Mill Products Division 600 Buncombe St. P.O. Box 3307 Greenville, SC 29602				×							
Con-Vel, Inc. 166 Quality Dr. Summerville, SC 29483				×							
Control Concepts Inc. 100 Park St. Putnam, CT 06260											×
Coupling Corporation of America 2424 S. Queen St. York, PA 17402									×		
Curtis Universal Joint Co. P.O. Box 38 Springfield, MA 01107				×							

	Mechanical element				Elastomeric element			Metallic element			
Manufacturer	Gear	Grid	Chain	U-Joint	Shear	Compression	Combined	Disc	Diaphragm	Misc.	Miniature
Dalton Gear Co. 212 Colfax Ave. Minneapolis, MN 55405			×								
Dana Corp./Spicer Universal Joint Division P.O. 955 Toledo, OH 43695				×							
Dayco Corp. 1 Prestige Place P.O. Box 1004 Dayton, OH 45401					×						
Diamond Chain Co. P.O. Box 7045 402 Kentucky Ave. Indianapolis, IN 46207			×								
Dodge/Reliance Electric P.O. Box 499 Greenville, SC 29602		×	×		×	×					
Falk Corp., Unit of Sundstrand Corp. Box 492 Milwaukee, WI 53201	×	×			×			×			

Fenner-Manheim 311 W. Stiegel St. Manheim, PA 17545											
FlexElement P.O. Box 750635 Houston, TX 77275								×			
Flexibox, Inc. 2407 Albright St. Houston, TX 77017								×			
Guardian Industries 3201-T Ohio St. P.O. Box 478 Michigan City, IN 46360	×				×	×					
Helical Products Co., Inc. P.O. Box 1069 Santa Maria, CA 93456										×	
Johnson Power, Ltd. 2530 Braga Dr. Broadview, IL 60153				×							
Kamatics Corp. 1335 Blue Hills Ave. Bloomfield, CT 6002								×			
Koeling-Morgan 15 Belmont St. Worcestor, MA 01605				×							

	Mechanical element				Elastomeric element			Metallic element			
Manufacturer	Gear	Grid	Chain	U-Joint	Shear	Compression	Combined	Disc	Diaphragm	Misc.	Miniature
KTR Corp. Box 9065 Michigan City, IN 46360					×						
Lo-Rez Vibration Control, Ltd. 186 West 8th Ave. Vancouver, B.C., Canada V5Y 1N2										×	
Lord Corp./Mechanical Products Division 2000 W. Grandview Blvd. P.O. Box 10038 Erie, PA 16514					×	×					
Lovejoy 2655 Wisconsin Ave. Downers Grove, IL 60515	×				×	×	×				
Lucas Aerospace Power Transmission Corp. 211 Seward Avenue P.O. Box 457 Utica, NY 13503									×		
Magalloy Coupling Co. P.O. Box 455-T Alpena, MI 49707						×					

Maurey Mfg. Co. 2907 S. Wabash Ave. Chicago, IL 60616					×	×					
Mayr Corp. 4 North St. Waldwick, NJ 07463								×		×	
Morse Industrial/Emerson Power Transmission Corp. S. Aurora St. Ithaca, NY 14850					×	×					
Renolds/Ajax Inc. P.O. Box A Westfield, NY 14787	×			×							
Rexnord Corporation—Elastomeric Products Oper (Omega Couplings) 4701 W. Greenfield Ave. P.O. Box 2022 Milwaukee, WI 53201					×						
Rexnord Corporation—Thomas Coupling Oper. 304 Main Ave. Warren, PA 16365								×			
Ringfeder Corp. 165 Carver Ave. P.O. Box 691 Westwood, NJ 07675								×			

	Mechanical element				Elastomeric element			Metallic element			
Manufacturer	Gear	Grid	Chain	U-Joint	Shear	Compression	Combined	Disc	Diaphragm	Misc.	Miniature
Rockwell Automotive 2135 W. Maple Rd. Troy, MI 48084				×							
Servometer Corp. 501 Little Falls Rd. Cedar Grove, NJ 07009											×
Stock Drive/Sterling Instrument 2101 Jericho Tpke. Box 5416 New Hyde Park, NY 11040				×							
System Components, Inc. 1635 Stieve Dr. P.O. Box 129 South Haven, MI 49090	×							×			
The Indikon Company, Inc. 76 Coilidge Hill Rd. Watertown, MA 02172										×	
Torquemeters, Ltd. Ravensthorpe Northampton, England										×	
Twin Disc, Inc. 1328 Racine Racine, WI 53403				×							

Voith Transmission, Inc. 25 Winship Rd. York, PA 17402				×							
Walterscheild, Inc. 16 W .030-83rd St. Burr Ridge, IL 60521				×							
T.B. Wood's Sons 440 N. Fifth Ave. Chambersburg, PA 17201	×				×	×		×			
XTEK, Inc. 11451 Reading Rd. Cincinnati, OH 45241	×										
Zero-Max, Inc. 13200 Sixth Ave. N. Minneapolis, MN 55441								×		×	

This page intentionally left blank

Appendix B

Typical Installation Instructions

I. INSTRUCTIONS FOR GEAR COUPLINGS

Alignment and Installation Instructions

Purpose: The purpose of aligning equipment is to avoid transmission of unwanted stresses to bearings, shafts, couplings, etc.

How: By providing minimum angularity and offset of shaft axis at normal operating conditions (Figs. 1 and 2)

Why: To increase life of bearings, couplings, shafts and seals. To get at the root of serious malfunctions involving shutdowns and costly repairs.

When:

1. During installation, before grouting.
2. Immediately after initial operation.
3. When final operating conditions and final temperature are attained.
4. Seasonally
5. Whenever first symptoms of trouble occur — vibration, undue noise, sudden overheating of bearings.

Practical Considerations:

1. Verify shaft separation
2. Locate rotor in running position (for example, on sleeve bearing motors).
3. Anticipate thermal changes.
4. Read instructions and review drawings.

Tools:

1. Dial indicator with attaching device.
2. Feeler gauges.
3. Inside micrometer.
4. Outside micrometer.
5. Snap gauges.
6. Straight edge.

Angular Misalignment Measurement:

1. Measure at 4 points the space between the shaft ends (Fig. 3)
2. Rotate both shafts 180° and repeat.
3. Perform calculations for angle.

Offset Misalignment Measurement:

1. Rotate shaft A (with dial indicator mounted) and note readings of shaft B offset (Fig. 4).
2. Or use straight edge and feeler gauge (Fig. 5).

CAUTION: Misalignment at installation should not exceed 1/3 of rated catalog misalignment.

CAUTION: Rotating equipment is potentially dangerous and could cause injury or damage if not properly protected. Follow applicable codes and regulations.

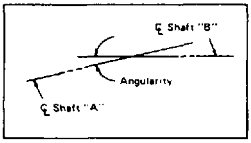


Figure 1 — Angularity is the acute angle formed at the intersection of the axes of the driving and the driven machine shafts. When shafts are exactly parallel, angular misalignment is zero; but vertical or horizontal displacement of axes may be present (See Fig. 2)

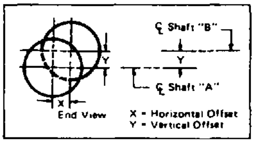


Figure 2 — Concentric alignment (also called offset alignment or parallel offset) is the relationship between the shaft axes in terms of vertical and horizontal displacements of the axis of one shaft from the axis of the other shaft.

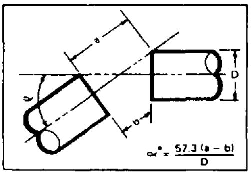


Figure 3 — To determine relative angular shaft-positions of driving and driven machines, measure at four points the space between the shaft ends. Choose the largest (a) and smallest dimension (b).

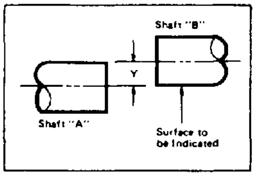


Figure 4 — To measure offsets with a dial indicator, attach the indicator to shaft "A", rotate shaft, and indicate to the periphery of shaft "B". To obtain actual displacements of shafts, divide dial indicator readings by 2.

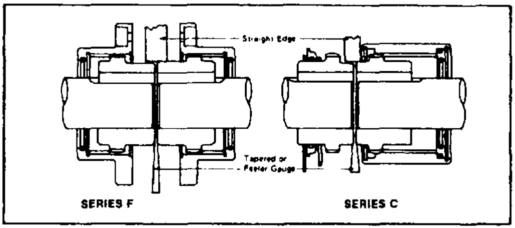
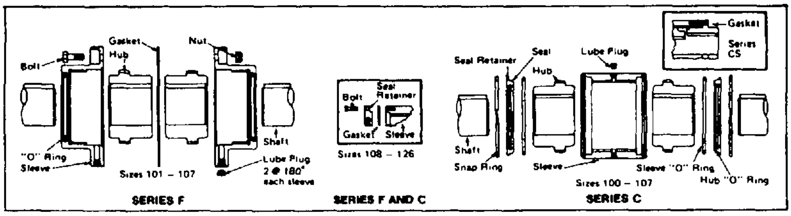


Figure 5 — Lay straight edge on one hub and measure gap between straight edge and other hub with feeler gauge. Measure at top, bottom, and both sides. Feeler gauge readings indicate actual displacements of shafts.

Installation and Lubrication Instructions



SERIES F

SERIES F AND C

SERIES C

Installation
Disassemble coupling and clean all parts. Follow the appropriate 6 steps below and you are ready to go! Installed and lubricated in accordance with the instructions, your Amerigear coupling is prepared for a life of dependable, trouble-free service

Series F Installation
Step 1. Lightly coat grease on "O" rings and insert "O" rings into grooves of sleeve (into grooves of seal retainer for sizes 108 and larger). Place sleeves (sizes 100-107) over shaft ends. For sizes 108 and larger, place only the seal retainer (with "O" rings inserted on shaft and check orientation to assure male pilot is toward shaft end. Care should be taken not to damage seal on shaft key seat.
Step 2. Check key fits and coat keys and keyways with oil resistant sealing compound (Permatex) to prevent leakage. Install size 101 through 107 hubs on shafts with long ends flush with shaft ends. Install size 100 hub on shaft with short end flush with shaft end. For shrink fits, apply heat to hubs uniformly, preferably submerged in oil not exceeding 350°F. Do not allow "O" ring seals to contact heated hubs. For sizes 108 and larger, place retainer gaskets and sleeves over hubs and onto shafts.
Step 3. Align shafts allowing clearance as per tabulation or in accordance with Dimension "D" from Engineering Data. Check gap with taper or feeler gauge at 90° points and align hubs with straight edge at 90° points.
Step 4. After thoroughly coating hub and sleeve teeth with lubricant, slip sleeves onto hubs, carefully engaging teeth (do not damage seal surface). Place sleeve gasket between sleeves and align bolt holes.
Step 5. Secure sleeves, using care to tighten fasteners uniformly. See tabulation "Flange Bolt Tightening Torque". For sizes 108 and larger bolt seal retainers to sleeves.
Step 6. Remove both Dryseal lube plugs and add grease in sufficient amount to overflow with lubricant holes in horizontal position. Install lube plugs using Permatex No. 2 for sealing and seat securely. For limited end float variation, Series FI and FP, lube both halves of coupling separately. For vertical couplings, remove one Dryseal plug in each coupling half and carefully insert thin probe under upper hub seal to assure complete filling.

Series C Installation
Step 1. For sizes 100-107 place snap ring, seal retainer (with "O" ring seated in retainer groove) and sleeve "O" ring on each shaft. For sizes 108 and larger, place seal retainer (with "O" ring inserted) and gasket over shaft and check orientation to assure male pilot is toward shaft end. For Series CS install gasket between rigid hub flange and sleeve.
Step 2. Check key fits and coat keys and keyways with oil resistant compound to prevent leakage. Install hubs on shafts with short ends flush with shaft ends. For shrink fits, apply heat to hubs uniformly, preferably submerged in oil not exceeding 350°F. Do not allow "O" rings to contact heated hubs.
Step 3. Slip sleeve over hub mounted on longest shaft.
Step 4. Align shafts allowing clearance as per tabulation or from Engineering Data, Dimension "D". Check gap with taper or feeler gauge at 90° intervals Also align hubs with straight edge at 90° points.
Step 5. Pack hub and sleeve teeth with grease. Force grease into shaft gap. Lightly coat grease on "O" rings. Slide sleeve over hubs to center position. Remove Dryseal lube plugs* and add lubricant in sufficient amount to overflow capacity. For vertical application remove one Dryseal plug and carefully insert probe between upper hub seal to assure coupling is full of lubricant.
Step 6. For sizes 100-107 install sleeve "O" rings in sleeve counterbores — then press seal retainer assembly in place. Use fingertips or blunt tool. Seat snap rings in grooves, using a winding motion. Recheck to assure snap rings are positively seated. For sizes 108 and larger, bolt seal end plates to sleeve.

	HUB SEPARATION			FLANGE BOLT TIGHTENING TORQUE FT. LBS.*	
SIZE	F & C	FS	CS	F Exposed	F Shrouded
100	.125	.078	.125	10	10
101	.125	.078	.125	10	10
101¼	.125	.078	.125	10	10
101½	.125	.156	125	29	32
102	.125	.156	.125	63	32
102½	.188	.188	.188	125	69
103	.188	.188	.188	125	69
103½	.250	.219	.250	210	133
104	.250	.312	.250	210	133
104½	.312	.344	.312	210	133
105	.312	.344	.312	313	232
105½	.312	.344	.312	313	232
106	.312	.406	.312	313	340
107	.375	.500	.375	440	476
108	.375	.500	–		
109	.500	.562	–		
110	.500	.625	–		
111	.500	.625	–		
112	.500	.625	–		
113	750	.750	–		
114	.750	.750	–		
116	.750	.750	–		

*Tightening torque based on unlubricated threads, if threads are lubricated derate torque to 75% of above values.

Maintenance and Lubrication

LUBRICANTS

LUBRICANT MANUFACTURER	GENERAL	MOIST OR WET	HIGH TORQUE	160° - 300°F	CLASS III
American Lubricants Co	Alubco Bison 1650	(same)	(same)	(same)	(same)
Atlantic Richfield Co.	Arco MP	Arco MP	Arco EP or Moly D	Dominion H2	Dominion H3
Amoco	Amolith #2	Amolith #2	Amolith #2	Rykon EP-2	Amoco CPLG Grease
Chevron USA, Inc.	Duralith EP-2	Duralith EP-2	Duralith EP-2	Duralith EP-2	NL Gear Compound 460
Cities Service Oil Co.	Citgo AP or HEP-2	AP or HEP-2	AP or HEP-2	AP or HEP-2	Citgo AP or EP Compound 130
Continental Oil Co.	Super Sta Grease	HD Calcium Grease	HD Calcium	HD Calcium	HD Calcium, Transmission Oil No. 140
Far Best	Molyvis ST-200	(same)	(same)	(same)	--
Fiske Bros. Refining Co	Lubriplate 630-AA	Lubriplate 630-AA	Lubriplate 630-AA	Lubriplate 1200-2	Lubriplate No. 8
Gulf Oil Co.	Gulfcrown EP #2	Gulfcrown EP #2	Gulfcrown EP #2	Hi-Temp Grease	Precision No. 3
Exxon Co	Pen-O-Lead EP 350	Rolubricant EP-300	Rolubricant EP-350	Unirex N2	Unirex N2 or Nuto No. 146
Kendall Refining Co.	Kenlube L-421 or Waverly Torque Lube A	L-421 or Torque Lube A	L-421 or Torque Lube A	L-421 or Torque Lube A	L-427
Mobil Oil Co.	Mobilux EP-0	Mobilux EP-0	Mobil Temp 78	Mobil Temp 78	Mobil No. 28
Pennzoil Co.	Pennlith 711 or 712	Pennlith 711 or 712	Pennlith 711 or 712	Pennlith 712 or Bearing Lube 706	Hi Speed Pennlith 712 or Bearing Lube 706
Suntech, Inc.	Sunaplex 991 EP or Prestige 741 EP	(same)	(same)	(same)	--
Syn-Tech	3913-G1	(same)	(same)	(same)	(same)
Tenneco Chemicals, Inc.	Anderol 786	(same)	(same)	(same)	(same)
Texaco, Inc.	Multifak EP-2	Multifak EP-2	Multifak EP-2	Thermatex EP-2	Thermatex EP-2
Union Oil Co. of Calif.	UNOBA EP-2	UNOBA EP-2	UNOBA EP-2	UNOBA EP-2	MP Gear Lube 140

For low temp. (-65°), Aeroshell #22 by Shell Oil Co., Anderol 793 by Tenneco Chemicals, Inc. & Mobil Grease #28 by Mobil Oil Co.

LUBRICANT QUANTITIES

Coupling Size	LUBRICATION			
	SERIES "F"		SERIES "C"	
	Wt. Lbs.	Vol. Qts.	Wt. Lbs.	Vol. Qts.
100	.020	.010	.016	010
101	.046	.025	.036	.020
101¼	.06	.033	.046	.026
101½	.18	.10	.09	.05
102	.34	.19	.18	.10
102½	.45	.25	.30	.17
103	.79	.44	.45	25
103½	1.0	.56	.57	31
104	1.7	.94	1.1	.63
104½	2.0	1.1	1.2	.67
105	3.6	2.0	1.8	1.0
105½	5.0	2.7	2.3	1.2
106	5.9	3.2	2.7	1.6
107	10.9	6.0	6.0	3.3
108	16.6	9.2	8.0	4.4
109	20.3	11.2	10.9	6.0
110	25.4	14.0	13.8	7.6
111	38.4	21.2	22.5	12.4
112	43.5	24	25.4	14.0
113	52.2	28.8	31.2	17.2
114	68.9	38	42.8	23.6
115	77.5	42.8	49.3	27.2

Maintenance — The Amerigear Coupling requires a minimum of maintenance. Nevertheless, to ensure a trouble-free life a few checks and proper lubrication should be performed at regular intervals.

Zurn suggests that the maximum interval between checks and relube be one year. This is only a guide, and the actual interval should be in accordance with good operating practices for application.

To disassemble Series F remove flange fasteners, separate sleeves, slide sleeves over hubs, clean out old lubricant, and inspect seals and gear teeth. Reassemble, starting with Step 3 under Series F installation instructions on the previous page.

To disassemble Series C, remove one snap ring, slide sleeves off hubs, clean out old lubricant and inspect seals and gear teeth. Reassemble, starting at Step No. 4 under Series C installation instructions on the previous page.

If proper alignment of shafts is assured and it is not practical to disassemble coupling, remove both lube plugs and add grease in sufficient amount to overflow with lubricant holes in horizontal position. Recommended lubricants and quantities are listed on this page.

NOTE: Sizes 100 and 101 Series C are supplied without lube plugs — lubricate per Series C Step No. 5

The lubricants listed above are recommended by the lubricant manufacturers for the indicated conditions. This list is solely for our customers' convenience and does not constitute an endorsement. The listing is not intended to be complete nor necessarily current due to continuous research and improvement by the various manufacturers.

Series F, FM, FA use quantities as recommended.

Series FS, FMS, FAS use one-half the quantities recommended.

Series C, CM, CA use quantities as shown.

Series CS, CMS, CAS use one-half the quantities recommended.

*Series F Class III use quantities as recommended for Series F but limited to the greases shown in Class III column above or the following oils.

Citgo EP Compound 130 by Cities Service, Nuto No. 146 by Exxon, Lubriplate No. 8 by Fiske Bros.; Transmission Oil No. 140 by Continental.

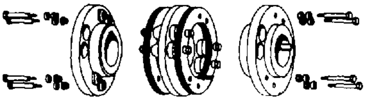
II. INSTRUCTIONS FOR DISC COUPLINGS

INSTALLATION AND ALIGNMENT
FLEXIBLE DISC COUPLINGS

INSTALLATION

Reasonable care in initial assembly and aligning will permit couplings to operate to full capacity in correcting future misalignments.

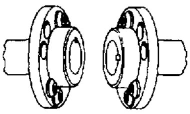
CAUTION
All rotating power transmission producrs are potentially dangerous and must be properly guarded.



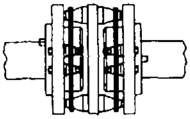
1. Coupling is shipped with center member assembled as shown above. If it is necessary to disassemble the coupling completely, tie a string or wire thru the bolt holes to maintain the dialing (alternating grain of the individual discs). Be careful to note the arrangement of parts.

2. Inspect both shafts and hub bores, making sure they are free from burrs. Be sure the keys fit the shafts and hubs properly.

3. Mount hubs on shafts. If hub is bored for an interference fit, they should be heated in water, oil or with soft open flame and quickly positioned on the shaft. Do not spot-heat hub as it may cause distortion.



4. Move equipment to be connected into position, making sure that the distance between hubs is equal to the "C" dimension given in coupling dimensional tables ("L" dimension on floating shaft coupling types.)



5. Reassemble coupling, keeping the proper order of parts as noted in Step 1. Refer to drawings on coupling listing pages.

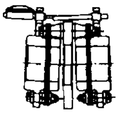
Note that coupling is in alignment.

NOTE: Floating shaft couplings should be mounted close to bearings of connected units to minimize overhang. Cooling tower and other right angle drives—dowel right angle gear boxes to their bases after couplings have been aligned. Right angle gear boxes tend to creep in a counter-rotational direction, which causes severe coupling misalignment and may result in premature failures.

ALIGNMENT

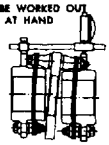
INDICATOR METHOD

INDICATOR MOUNTING TO BE WORKED OUT TO SUIT CONDITIONS AT HAND



CHECK FOR ANGULAR MISALIGNMENT

Dial indicator measures maximum longitudinal variation in hub spacing through 360° rotation.

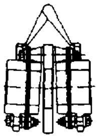


CHECK FOR PARALLEL MISALIGNMENT

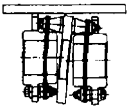
Dial indicator measures displacement of one shaft center line from the other.

1. Attach dial indicator to hub, as with a hose clamp; rotate coupling 360° to locate point of minimum reading on dial; then rotate body or face of indicator so that zero reading lines up with pointer.
2. Rotate coupling 360°. Watch indicator for misalignment reading.
3. Driver and driven units will be lined up when dial indicator reading comes within maximum allowable variation of .002" for each inch of flange diameter; ie. a coupling with a 4-inch diameter flange has a maximum variation of .008 inches in indicator reading.
4. Reset pointer to zero and repeat above operations 1 and 2 when either driven unit or driver is moved during aligning trials.
5. Check for parallel misalignment as shown. Move or shim units so that parallel misalignment is at a minimum.
6. Coupling should be rotated several revolutions to make sure no "end-wise creep" in connected shafts is measured.
7. Tighten all locknuts.
8. Re-check and tighten all locknuts after several hours of operation.

CALIPER AND STRAIGHT EDGE METHOD



CHECK FOR ANGULAR MISALIGNMENT



CHECK FOR PARALLEL MISALIGNMENT

1. Use calipers to check the gap between hubs. Gap should be the same at all points around the hub.
2. Place straight edge on the rims at the top and sides. When the coupling is in alignment the straight edge should rest evenly and both disc pack assemblies should be in a perfect plane at right angles to the straight edge.
3. Tighten all locknuts.
4. After several hours of operation recheck gap between hubs, and recheck tightness of all locknuts.
5. When operating at full speed, disc packs should have a clearly defined appearance—not blurred when viewed from top and side.

III. INSTRUCTIONS FOR ELASTOMERIC COUPLINGS

SURE-FLEX INSTALLATION INSTRUCTIONS

Sure-Flex flanges (outer metallic parts) and sleeves (inner elastomeric members) come in many sizes and types.

First, determine the size and type of components being used. Remove all components from their boxes, and loosely assemble the coupling on any convenient surface.

All rubber sleeves (EPDM and Neoprene) have the same ratings for a given size and may be used interchangeably. Hytrel sleeves, however, have completely different ratings. Rubber sleeves should not be substituted for Hytrel, or Hytrel for rubber.

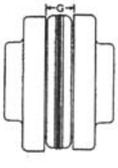
(Do not attempt to install the wire ring on the two-piece E or N sleeve at this time.)

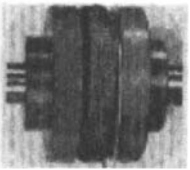
1 Inspect all coupling components and remove any protective coatings or lubricants from bores, mating surfaces and fasteners. Remove any existing burrs, etc. from the shafts.

2 Slide one coupling flange onto each shaft, using snug-fitting keys where required. With the Type B or C flanges, it may be necessary to expand the bore by wedging a screwdriver into the saw cut.

3 Position the flanges on the shafts to achieve the G dimension shown in table at right. It is usually best to have an equal length of shaft extending into each flange. Tighten one flange in its final position. Slide the other far enough away to install the sleeve. With a two-piece sleeve, do not move the wire ring to its final position; but allow it to hang loosely in the groove adjacent to the teeth.

4 Slide the loose flange on the shaft until the sleeve is completely seated in the teeth of each flange. (The "G" dimension is for reference and not critical.) Secure the flange to the shaft. (Note: When using sleeve bearing electric motors, the armature must be at its electrical center when determining sleeve engagement and flange location.)

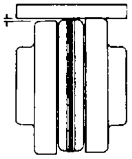




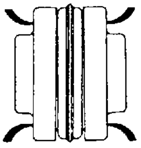
Different coupling sleeves require different degrees of alignment precision. Locate the alignment values for your sleeve size and type in table at right.

5 Check parallel alignment by placing a straightedge across the two coupling flanges and measuring the maximum offset at various points around the periphery of the coupling. DO NOT rotate the coupling. If the maximum offset exceeds the figure shown under "Parallel" in table below, realign the coupling.

6 Check angular alignment with a micrometer or caliper. Measure from the outside of one flange to the outside of the other at intervals around the periphery of the coupling. Determine the maximum and minimum dimensions. DO NOT rotate the coupling. The difference between the maximum and minimum must not exceed the figure given under "Angular" in table below. If a correction is necessary, be sure to recheck the parallel alignment.



Parallel



Angular

MAXIMUM ALLOWABLE MISALIGNMENT

(Dimensions in inches)

Sleeve Size	G Dimension	Types JE & JN		Types E & N		Type H	
		Parallel	Angular	Parallel	Angular	Parallel	Angular
3	⅜	010	.035	....	....	....	....
4	⅝	010	.043	.010	.043	....	....
5	¾	.015	.056	.015	.056	....	....
6	⅞	.015	.070	.015	070	010	016
7	1	.020	.081	.020	.081	012	.020
8	1⅛	.020	.094	020	094	015	025
9	1⁷⁄₁₆	....	....	025	.109	.017	028
10	1⅝	....	....	.025	.128	.020	.032
11	1⅞	....	....	.032	.151	022	.037
12	2⁵⁄₁₆	....	....	.032	175	025	042
13	2¹¹⁄₁₆	....	....	.040	195	.030	.050
14	3¼	....	....	045	.242	.035	.060
16	4¾	....	....	.062	.330	....	....

Note: Values shown above apply if the actual torque transmitted is more than ¼ the coupling rating. For lesser torque, reduce the above values by ½.

7 If the coupling employs the two-piece sleeve with the wire ring, force the ring into its groove in the center of the sleeve. It may be necessary to pry the ring into position with a blunt screwdriver.

8 Check safety codes, and install protective guards or shields as required.

Caution: Coupling sleeves may be thrown from the assembly when subjected to a severe shock load.

References

American Gear Manufacturers Association. Load classification and service factors for flexible couplings. AGMA 922-A96, 1996.

American Gear Manufacturers Association. Nomenclature for flexible couplings. AGMA 510, 1984.

American Gear Manufacturers Association. Flexible couplings—lubrication. AGMA 9001-B97, 1997.

American Gear Manufacturers Association. Bore and keyways for flexible couplings (inch series). AGMA 9002-A86, 1986.

American Gear Manufacturers Association. Flexible couplings—potential unbalance classification. AGMA 9000-C90, 1990.

American Gear Manufacturers Association. Metric dimensions for gear coupling flanges. AGMA 561.01, 1978.

American Gear Manufacturers Association. Flexible couplings—keyless fits. AGMA 9003-A91, 1991.

American Petroleum Institute. Pumps for refinery service. API 610, American Petroleum Institute, 1990, 1998.

American Petroleum Institute. Special-purpose couplings for refinery service. API 671, 2nd and 3rd ed., American Petroleum Institute, 1990, 1998.

ANSI. Balance quality of rotating rigid bodies. ANSI S2 19-1989, 1989.

Barlett Hayward Company. Fast's Self-Alignment Couplings Catalogue, 1930.

Boylan, W. Marine Application of Dental Couplings. Society of Naval Architects and Marine Engineers, 1966.

Broersma I. G. Couplings and Bearings. H. Stam N.V., 1968.

Calistrat, M., and R. Munyon. Design of Coupling Enclosures. Lecture—Turbomachinery, 14th Symposium, 1985.

Calistrat, M. Extended gear coupling life (parts 1 and 2). Hydrocarbon Processing, 1978.

Calistrat, M. Flexible Coupling Installation. Koppers, Baltimore, 1968.

Calistrat, M. Friction Between High Speed Gear Coupling Teeth. ASME 80-C2/DET 5.

Calistrat, M. Grease Separation Under Centrifugal Forces. Koppers 3924.

Calistrat, M. Flexible Couplings: Their Design, Selection, and Use. Caroline Publishing, 1994.

Calistrat, M. Shaft lubrication. Journal of the American Society of Lubrication Engineers, 1980, pp. 9–15.

Calistrat, M. Wear and lubrication. Mechanical Engineering, 1975.

Calistrat, M. Sludge accumulation in continuously lubricated couplings. Koppers 1645.

Calistrat, M. Hydraulically Fitted Hubs: Theory and Practice. Lecture—Turbomachinery, 9th Symposium, 1980.

Carter, D., M. Garvey, and J. Corcoran. Baffling and Temperature Predictions of Coupling Enclosures. Lecture—Turbomachinery, 23rd Symposium, 1994.

Chase, T. W. Maintenance of gear-type couplings. Iron and Steel Engineer, 1964, pp. 72–84.

Chironis, N. Gear Design and Application. McGraw-Hill, New York, 1967.

Condon, W. Application of Universal Joints to Construction and Industrial Machinery. Society of Automotive Engineers, 1961.

Dodd, R. Total Alignment. Petroleum Publishing Co., 1975.

Eschleman, R. Combating vibration with mechanical couplings. Machine Design, 1980.

Faires, V. M. Design of Machine Elements, 4th ed. Macmillan, New York, 1965.

Firth, D. Couplings for steel mill drives. Iron and Steel Engineer, 1963.

Genshelmer, J. How to design flexible couplings. Machine Design, 1961.

Gibbons. Diaphragm Couplings in Turbomachinery. Machinery Vibration Institute, 1979.

Gibbons, J. Mancuso, and R. Munyon. Tutorial on Couplings. Turbomachinery, 18th Symposium, 1989.

IRD Mechanalysis. Audio-Visual Customer Training, Instruction Manual, 1975.

ISO. Flexible couplings for mechanical power transmission—special purpose applications. ISO DIS 10441.

ISO. Flexible couplings for mechanical power transmission—general purpose applications. ISO DIS 14691.

ISO. Balance quality of rotating rigid bodies. ISO 1940/1, 1986.

Wilson, W. K. Practical Solutions of Torsional Vibration Problems. Chapman & Hall, 1968.

Lord Corporation. Understanding Torsional Vibration. Design Monograph DM 1107, 1978.

Lovejoy, Inc. Power Transmission in the Twentieth Century: An Overview. Downers Grove, IL, 1988.

Lovejoy, Inc. Preventing Unnecessary Failure of Power Transmission Components. LJ-702, Downers Grove, IL, 1985.

Mancuso, J. Retrofitting gear couplings with diaphragm couplings. Hydrocarbon Processing, 1988.

Mancuso, J. Let's try to really understand coupling balance. ASME, 1996.

Mancuso, J. Manufacturers world of coupling potential unbalance. Lecture—Turbomachinery, 13th Symposium, 1984.

Mancuso, J. General purpose vs. special purpose couplings. Tutorial—Turbomachinery, 23rd Symposium, 1994.

Mancuso, J., J. J. Zilberman, J. S. Corcoran, and D'Ercole. Flexible-element couplings: how safe is safe? Hydrocarbon Processing, December, 1994.

Mil-C-23233. For Couplings for Propulsion Units, Auxiliary Turbines and Line Shafts Naval Shipboard.

Muster, J., and B. Flores. Balancing Criteria and Their Relationship to Current American Practices. ASME Paper 69-Vibo-60.

Neale, M., et al. Proceedings of the International Conference on Flexible Couplings for High Power and Speeds. 1977.

Peterson, R. Stress Concentration Factors. Wiley-Interscience, New York, 1974.

Piotrowski, J. Shaft Alignment Handbook, 2nd ed. Marcel Dekker, New York, 1995.

Poole Foundry and Machinery Co. Catalog. 1958.

Potgietier, F. Cardan universal joints applied to steel mill drives. Iron and Steel Engineer, 1969, pp. 73–78.

Roots, F. Shaft Couplings. U.S. Patent 349,365, 1886.

Rothfuss, N. Design and Application of Flexible Diaphragm Couplings to Industrial-Marine Gas Turbine. ASME 73-GT-75.

Rothfuss, N. Design criteria and tables for selecting high speed power couplings. Product Engineering, 1963.

Schwerdlin, H. Reaction forces in elastomeric couplings. Machine Design, 1979.

Society of Automotive Engineers. Universal Joint and Driveshaft Design Manual. AE-7.SAE, 1979.

South, D., and J. Mancuso. Mechanical Power Transmission Components. Marcel Dekker, New York, 1994.

Texaco, Inc. Lubrication of Flexible Couplings, Vols. 1 and 2.

Thomas Flexible Coupling Co. Thomas Flexible Couplings Catalog. 1942.

Wellman Bibby Co. The Bibby Resilient Coupling.

Wolford, C. Retrofitting turbomachinery with high performance flexible dry couplings. Lecture—Turbomachinery, 19th Symposium.

Wood's, TB, Sons Company. Home Study Course on Couplings, 1970.

Wright, J. Tracking down the cause of coupling failure. Machine Design, 1977.

Wright, J. Transient Torsional Vibration in Synchronous Motor Drives. ASME Paper 75-DE-15, 1975.

Zurn Industries, Inc. The evolution of environmentalism, 1973.

Zurn, F. Crowned Tooth Gear Type Couplings. AISE, 1957.

Index

AGMA
  balance classes, 134,138–139, 141
  keys and keyways, 157–163
  shaft fit, 153, 156
Alignment, 193–216
  cold method, 197–211
  hot method, 211–213
  importance of, 195–196
  laser, 210–216
  tolerance guide, 194
  vibration signatures, 213–215, 217–221
American Gear Manufacturers Association (*see* AGMA)
American Petroleum Institute (API)
  balance, 136–137, 139
  safety factors, 87
Angular misalignment, 98, 200–202
Assembly, 215–240
  elastomeric, 224, 568
  flexible metallic element , 223–224, 567
  gear couplings, 227–240, 564–566
  lubricated coupling, 216, 220, 222–223
Assembly balance, 128, 130–132
Attachment methods for couplings, 154–155

Axial force
  gear coupling, 283–284
  general equation, 101
Axial natural frequency, 445–448
Axial stiffness
  flexible element, 445–448
  linear vs. nonlinear, 445–448
Axial travel, 25–26, 70–71

Balancing, 118–152
  AGMA classes, 134
  arbitrary limit, 134, 136–138
    micro-inch limit, 137–138
    oz-in. limit, 136–137
  assembly balance
    mandrel, 128, 130–131
    mandrelless, 128, 129, 132
  basics of, 118–124
  classifications, 138–139
    weight vs. speed, 139, 141
  component, 127–133,129–130
  example selection, 140–152
    summary of, 140
  field, 128, 131, 133
  fundamentals, 119–122
  how to, 126–128
  levels of, 138–139
  mandrelless, 128, 131, 132
  mandrels, 122, 123, 128, 131
  manufacturing, tolerance effect, 127, 140
  potential, 124, 134, 136–138
  residual, 124, 134, 136–138
  single-plane, 120–121
  two-plane, 121
  types, 128–133
  U-joint, 132
  unbalance
    causes of, 125–126
    force, 124
[Balancing]
  [unbalance]
  when to, 138–139
  why, 124–125
Bending moment
  gear coupling, 284–287
  general equation, 100–101
Bore information, 79
Bolt grades, 110
Bushings, 164, 182–183
  loosening of, 245–246

Capacities, 70–71
  axial travel, 70–71
  bore, 70–71
  misalignment, 70–71
  speed, 70–71
  torque, 70–71
Cardon joint, 2
Centrifugal force, effect on
  diaphragm stresses, 441–442
  lubrication, 233–240
  speed, 90
Chain coupling, 354–361
  applications, 354, 357
  covers, 359
  design and construction, 355
  first, 3
  how they work, 354, 356
  lubrication, 227–228
  nylon, 40, 355, 358
  steel, 39, 354, 358
  selection, 361
  types, 354–355, 358
  typical sizes, 360
Characteristics of various types, 68–69, 72–73
Clearance shaft
  installation, 179–181
  standards, 156

Coefficient of friction
  bolt threads, 108–109
  flanges (between), 106–107
  gear coupling teeth, 283–284
  shaft to hub, 115
Component balancing, 126–128, 129–130, 150–152
Crowned tooth gear coupling, 38, 264–265, 266–267, 282–283, 290, 292–293, 298, 300, 322, 325

Design equations and parameters, 101–118
Diaphragm coupling, 475–487
  application, 480–481
  construction, 480
  failure mode, 484–487
  first used, 3, 11, 16
  installation, 223–224
  high performance, 19, 64–66
  multiple convoluted, 66, 441–444, 447, 476, 480, 483–484, 486
  multiple straight, 64, 66. 482, 486
  one-piece type, 19, 65, 475–478
  reduced moment, 479
  tapered contoured, 64–65, 475–480. 482–485
  types, 64–66, 480–484
Disassembly of couplings, 240–246
Disc couplings, 462–475
  application, 465–467
  circular disc, 50, 463–464, 468
  construction, 465
  failure mode, 468–470, 473–475
  forms of disc, 462–465
  frame type, 64, 472
  high performance, 18, 62–63
[Disc couplings]
  installation, 223, 567
  link type, 51. 62, 464, 471–472
  pump type, 51
  square and miscellaneous shapes, 50–51, 463–464, 470
  types, 62–64, 467–472

Earliest couplings 1–9
Elastomeric couplings, 403–438
  differences, compression, and shear type, 403–405
  failure modes, 417–423
  how they work, 423–438
  installation, 224, 568
  material chemical resistance, 422–423
  torsional considerations, 409–417
Elastomeric compression couplings, 41–49, 403–405, 423–430
  block type, 20, 48, 61, 423–427
  donut or ring type, 47, 428–429
  jaw type, 48, 423–426
  pin and bushing, 49, 426–427
  types, 43–45, 47–49
Elastomeric shear couplings, 41–43, 45–47, 430–438
  bonded type, 60, 435
  donut type, 46–47, 436–438
  tire type, 45–46, 430–435
  types, 45–47, 50
End movement, 25–26

Failures
  diaphragm couplings, 484–486
  disc couplings, 473–475

[Failures]
  elastomeric couplings, 417–423
  equipment, 28, 406
  gear couplings, 27, 329–342
  grid couplings, 346
  U-joints, 368–371, 392–402
Fasteners for couplings
  assembly of, 224–227
  grades used, 110
  thread coefficient of friction, 109
Fatigue failure, 391
Flange connection, 171–175
  capacities and stresses, 106–112
Flange information, 79
Flanged rigid, 249, 250–252
Flattened bores, 163–164
Flexible couplings, advantages of, 23–29
Functional capacities, 70–71
Functions of couplings, 23–29

Gear couplings, 262–342
  affect on system
    axial force, 283–284
    bending moment, 284–287, 326–327
  application of, 265–266, 294–297
  backlash requirements, 263–265
  carburized, 274–277, 280, 317–321
  coefficient of friction, 283–284
    criteria for comparison, 280–283
    crowned tooth, 38, 263–265, 266–267, 290–293, 297–298, 300
    failure modes, 287–289, 329–342
[Gear couplings]
  [coefficient of friction]
    teeth, 287–289, 329–337
    other, 27, 158, 338–342
  flangeless, 304
  functionality, 257–272
  gas turbine type, 307
  general purpose, 257, 299–304
  ground gearing, 321
  high angle/high speed, 271–272, 312–314
  high angle/low speed, 268–271, 308–311
  high performance (high speed), 53–56, 306
  how they work, 263–265, 289–293
  how to evaluate, 280–283, 321–324
  induction hardened, 272–279, 317–320
  installation, 227–236, 564–566
  large, 295–296, 303
  lubrication, 227–240, 566
  materials and heat treatment, 272–280, 317–321
  material strength comparison, 318–319
  material wear resistance, 319–320
  miscellaneous applications, 314–316
  nitrided, 273–275, 279–280, 317–320
  percent of teeth in contact at various misalignment, 321
  PV criteria (ScV), 322
  spacer, 303
  spindle type, 11, 53–54, 268–271, 295, 308–311

[Gear couplings]
  [spindle type]
    comparison with u-joints, 59
    where used, 294
  special purpose, 267–268, 305–307
  standard gear coupling comparison, 301–302
  straight tooth form, 37, 266–267, 298–299
  tandem (floating shaft), 304
  tooth failures, 287–289, 329–337
  tooth patterns, 321, 328
  tooth stresses, 280–283, 321–324
    bending, 281
    compression, 282–283, 321–322
    shear, 281
  tooth sweep, 325
  torque capacity, 283, 323–324
  types, 37–38
    by application, 257–272
    by tooth form, 266, 298
  variables in design, 262–263
  where used, 265–266, 294–297
General purpose couplings, 34–52
  capacities, 70
  chain, 38–39
  characteristics, 72
  disc, 49–52
  elastomeric, 41–48
  gear, 37–38
  grid, 39–40
  miscellaneous, 52
  u-joints, 40–41
Grid couplings, 343–353
  accommodate misalignment, 343, 347
  affect on system, 345–346
[Grid couplings]
  application of, 344, 348–349
  design and construction, 344–345
  failure modes, 346
  horizontal cover type, 42, 344, 350
  selection, 352–353
  specials, 351–352
  torsional flexibility of, 343, 348
  types, 39–42, 350
  vertical split cover type, 41, 344, 350
  where used, 344, 348–349
Guards coupling, 457–462

Heat treatment gear couplings, 274–280
  importance, 275–276
History of couplings, 1–22
  early, 1–3
  1900–1930, 2, 4–7
  1930–1945, 4, 8–9
  1945–1960, 10–12
  1960–1985, 10, 13–14, 16–17
  1985–present, 15, 17–21
Hooke joint, 2
Hubs
  installation, 178–193
  keyed, 104–106, 112–113
  keyless, 113–118
  removal, 243–246
Hydraulic mounting, 169, 174

Industrial couplings, 29–30
Installation, 177–240
  alignment, 193–215
  disc coupling, 567
  do's & don'ts, 192–193
  elastomeric, 568
  fasteners, 224–227

[Installation]
  gear couplings, 564–566
  hubs, 177–193
  interference shafts, 178–182
  intermediate bushings, 182–183
  preparation for, 178–179
  rough bored couplings, 187–189
  tapered shafts, 184–187
Interface connections, 152–175
  flanges, 171–175
    face keys, 174–175
    shrouded vs. exposed bolt, 172–174
  keyless fit
    assembly, 165–171
    stresses, 113–118
  torque capacities, 114–118
  selection information, 79
  shaft connection, 152–175
    AGMA standard, 153, 156, 157–163
    bushings, 164
    clearance bores, 152–153
    flattened bores, 163
    interference fit, 153
    key and keywidth tolerances, 160
    keyed bores, 157–163
    keyless fit, 165–171
    keyless fit advance, 166–167
    setscrews, 153, 156
    splines, 164–165
    split hubs, 156–157
ISO balance requirements, 135
Interference shaft
  installation, 179–182
  standards, 165–169

Keyless fitted hubs, 165–171
  hub retention, 172–173
  o-ring arrangements, 168–169
[Keyless fitted hubs]
  oil injection methods, 169–170
  stress-relief grooves, 169, 171
Keys and keyways, 157–163
  AGMA standard, 160
  allowable stress limits, 106
  class of key fit, 159–160
  dimensions,160
  fillet radii, 163
  lead error, 161–162
  stresses, how to calculate, 104–106
  tolerances, 161–163
  types, 159
Key fit, 189–191
  failure, 158, 341–342

Laser alignment, 210–216
Lateral critical speed, 91–93
Lockup in gear couplings, 232–235, 287–289, 337–338
Lubrication, 227–240
  amount required, 232
  centrifugal force, effect on, 231–240
  methods and practices, 231–240
  grease, 237–240
  oil, 231–236
    anti-sludge, 234
    damless, 235
  types, oil vs. grease, 236–237

Maintenance, 240–243
Mandrels for balancing, 122, 123, 128, 131
Material limits, 93–97
  shrink fits, 180
Mechanically flexible couplings, 29–30,261–402
Metallic beam, 52

Metallic element, 439–487
  axial natural frequency, 445–448
  axial stiffness, 445–448
  heat generated and windage loss, 450–462
  lateral critical speed, 448–449
  stresses, type in membrane, 440–444
  system design considerations, 444–449
Miniature couplings, 29–30, 33–36
Misalignment, 25–26, 98–99
  capacity of coupling, 199–200
  definition of, 197
  how to calculate, 98–99, 198–199
  how to correct for, 200–210
    angular, 200–203
    offset, 200–203
  how to rough align, 193
  limits of equipment, 197, 199–200
  symptoms of, 196
  tolerance guide, 194
  types, 197–199
Miscellaneous couplings, 489–493
  pin and bushing, 52, 489–491
  Schmidt, 67, 489, 491
  slider block, 52–53, 489, 493
  spring
    tangential, 57, 491–492
    spiral, 35, 492–493
  types, 52–53, 64, 66–67
Modified Goodman diagram, 86

Offset misalignment,200–203
Oldham coupling, 493
ISO, 133–135

Pin and bushing coupling, 49, 426–427, 490–491
Pilots shrink fit effect, 191–192
Potential unbalance, 124, 134, 136–138
Power transmission, 23–24

Quill shaft coupling, 32, 248, 258, 260

Rating of couplings, 80–93
  correct, 89
  speed, 90–93
  torque, 80–90, 93
Reactions of couplings
  axial force, 101
  bending moment, 100–101
  unbalance force, 124
Residual unbalance, 124, 134, 136–138
Retrofitting nonlubricated couplings, 519–536
  example, 523–536
  guidelines
    weight-center of gravity, 523
    torsional stiffness, 524
  match the coupling, 521–533
  other considerations, 524–533
  responsibility, 520–521
  why change?, 519–520
Returning parts to service, 246
Ribbed rigid coupling, 31, 249, 254–255
Rigid couplings, 247–260
  application, 253, 255
  flanged rigid, 31, 247–252
  quill shaft, 32, 248, 258, 260
  ribbed rigid, 31, 249, 254–255
  sleeve rigid, 32, 253, 256–258
  types, 31–32, 253, 256–258

Safety factors, 81–87, 93–95
  Schmidt coupling, 67, 489, 491
Seals for couplings, 228–230, 311
Selection, 75–77
  capacities, 70–71
    characteristics, 72–73
    factors, 75–76
    information required, 77, 78–79
    responsibility, 75–80
    steps required, 76–77
    type for various torques and speeds, 68
  criteria, 82–84
Service factors, 81–83, 88–89
Set screws, 153, 156
Shaft stresses, 101–102
Shrink fit, 114–118
  heat mounting, 181
Sleeve rigid, 32, 253, 256–259
Slider block couplings, 52–53, 489, 493
Sludged-up gear coupling, 233, 288, 337
Special-purpose couplings, 52–57
  capacities, 71
  characteristics, 73
  diaphragm, 64–66
  disc, 62–63
  elastomeric, 57, 60–62
  gear type, 53–55
  miscellaneous, 64, 66–67
  u-joints, 55–57
Speed
  balancing requirements, 135, 141
  ratings based on, 90–93
  classes, 136–138
Splines, 103–104, 164–165
Split hubs, 156–157
Spring couplings
  spiral, 35, 492–493
  tangential, 57, 491–492
Straight toothed gear coupling, 37, 266–267, 298–299
Strength limits, 95–97
Stress equations
  elements (disc and diaphragms), 80–88, 440–444
  fasteners, 107–110
  flanges, 109, 111–112
  gear teeth, 280–283
  hubs, 112–113
  keys, 104–106
  shafting, 101–102
  splines, 103–104
Surface hardening, 276–280
System interactions, 77, 80

Tapered shafts, 184–187
  advance required, 184
  installation, 184–187
Torque, 98
Torque vs. speed, 68
Torquemeter coupling, 21, 495–508
  logistics, 496–497
  strain gage type, 497–499
  torsional deflection type, 501–508
    one channel, 506–508
    three channel, 503–506
    two-channel, 501–503
Torsional damping coupling, 508–519
  coupling design, 518–519
  damper couplings, 514–518
  machine analysis, 511–514
  machine vibration, 508–509
  torsional vibration, 509–511

Torsional stiffness, 99–100
Torsional vibration, 405–417
Transmissibility, 415–417
Types of couplings, 29–73

Unbalance force, 124
U-joints, 362–402
  application, 362, 373–377
  block type, 21, 58, 363, 372, 381–382, 396
  constant velocity, 57, 363, 379–380
  design and construction, 363–364, 381–386
  effect on system, 367–368, 391
  failure modes, 368–371, 392–402
  FEA yoke, 371
  flange types, 386
  industrial type, 55–59
  kinematic relationships, 364–367, 387–390
  lubrication, 227–228
[U-joints]
  mechanisms, 44, 362–363, 377–378
  problems, causes, and corrections, 399–402
  ratings, 367
  solid eye type, 17, 55–57, 363, 381, 384
  split eye type, 363, 372, 381, 383
  split yoke type, 363, 381, 384
  types, 40–41, 43–44, 55–59, 381–386
  vehicle type, 40–41, 43
Universal joints (*see* U-joints)
Urethane tire coupling, 45,

Weight (component), how to calculate, 99–100
Worm tracking (gear couplings), 287–289, 336
WR^2 (flywheel effect), how to calculate, 99–100

KOΛXO3
2/18/07